Attorney Docket No. AVID.19

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

APPLICANT: LUMPKIN	)	EXAMINER:
SERIAL NO.: 10/679,128	)	ART UNIT: 3723
FILED: OCTOBER 3, 2003	)	CONFIRMATION NO.: 2259
TITLE: SYMMETRIC CLAMP STRUCTURE	)	

Mail Stop: APPEAL BRIEF
Board of Patent Appeals and Interferences
P.O. Box 1450
Alexandria, Virginia 22313-1450

APPEAL BRIEF

Dear Sir:

In regard to the above referenced application, Appellant submits this Appeal Brief.

I. REAL PARTY IN INTEREST

The real party in interest is SRAM Corporation. SRAM Corporation's right to take action in the subject application was established by virtue of the following chain of title:

1. An Assignment from the inventor to Avid, LLC recorded at Reel 104582, Frame 0356.

2. An Assignment from Avid, LLC to SRAM Corporation recorded at Reel 014491, Frame 0358.

II. RELATED APPEALS AND INTERFERENCES

The undersigned legal representative of Appellant hereby confirms that there are no known appeals or interferences relating to the present application, or any parent application, which will directly affect or be directly affected by or have a bearing on the Board's decision in the pending appeal.

III. STATUS OF THE CLAIMS

Claims 1-17 are pending in the application. No claims have been allowed. Claims 1-17 stand rejected under a final Office Action mailed November 16, 2007 for the following reasons:

Claim 10 stands rejected under 35 USC § 112, second paragraph, as being indefinite for failing to particularly point out and distinctly claim the subject matter which Applicant regards as the invention.

Claims 1, 4, 6, 8-10 and 14-16 stand rejected under 35 U.S.C. § 102(b) as being anticipated by Nielsen, U.S. Patent No. 6,186,027. Claim 7 appears to also be rejected as anticipated by Nielsen (see p. 5 of the final Office Action).

Claims 2 and 3 stand rejected under 35 U.S.C. § 103(a) as being unpatentable over Nielsen, U.S. Patent No. 6,186,027, in view of Steinbock, U.S. Patent No. 6,381,827.

Claims 2 and 3 stand rejected under 35 U.S.C. § 103(a) as being unpatentable over Gelbein, U.S. Patent No. 5,584,210, in view of Nielsen, U.S. Patent No. 6,186,027, and further in view of Steinbock, U.S. Patent No. 6,381,827.

Claims 1, 4, 5, 11 and 17 stand rejected under 35 U.S.C. § 103(a) as being unpatentable over Gelbein, U.S. Patent No. 5,584,210, in view of Nielsen, U.S. Patent No. 6,186,027.

Claims 12 and 13 stand rejected under 35 U.S.C. § 103(a) as being unpatentable over Gelbein, U.S. Patent No. 5,584,210, in view of Nielsen, U.S. Patent No. 6,186,027, and further in view of Steinbock, U.S. Patent No. 6,381,827.

Each rejection of each of claims 1-17 is being appealed.

IV. STATUS OF THE AMENDMENTS

No amendment has been filed subsequent to the final rejection. Claims 1-17 are pending as amended by Applicant in the Response to Non-Compliant Amendment filed September 5, 2007, correcting various informalities with respect to the Amendment and Remarks filed June 6, 2007 in response to the Examiner's Office Action mailed March 9, 2007. The claims 1-17 set forth in Section VIII accurately reflect the claims pending.

V. SUMMARY OF CLAIMED SUBJECT MATTER

The claims are generally directed a clamp structure, a bicycle brake lever including the clamp structure, a method of attaching a clamp to a frame with the clamp structure and a method of manufacturing the clamp structure. No claim includes means plus function elements as permitted by 35 USC § 112, paragraph six.

Claim 1 as currently pending recites a clamp structure comprising a first arm 14 having a distal end defining a first threaded through bore18 and a second arm 16 having a distal end defining a second threaded through bore 20, wherein the first threaded through bore 18 and the second threaded through 20 bore are essentially coaxial. A screw 22 comprises a head 26 and a shank 24, with the head 26 being at one end of the shank and the shank having a threaded portion 28 at a second end opposite the first end. The screw further includes a clearance portion 30 between the threaded portion 28 and the head 26. (Fig. 3; specification p. 5, lines 19-28.) The screw 22 is configured so that with a threaded engagement between the threaded portion of the shank 28 and *either* of the first threaded through bore 18 of the first arm 14 or the second threaded through bore 20 of the second arm 16 and the head 26 abutting the other of the first and second arms opposite the threaded engagement, the clearance portion resides within the other of the first and second threaded through bores. (Figs. 4A and 4B; specification p. 6, lines 3-9.)

The structure recited in claim 1 provides a clamp wherein the screw can be inserted into either the first or second through bores of the first and second arms, respectively, and still perform a clamping function. This structure has the advantage of allowing the clamp to function in the event either of the first or second threaded through bores becomes stripped simply by inserting the screw into the opposite through bore.

Claims 2 and 3 further recite limitations directed to the diameter and length of the clearance portion 30 of the screw 22 relative to each of the threaded bores. These relationships are described at page 5, line 28 – page 6, line 2, and illustrated in Figs. 4A and 4B.

Claims 4 and 5 recite the clamp structure connected to a bicycle component and more particularly the bicycle component being a brake lever. See Figs. 1 and 2.

Independent claim 6, which is directed to a method of attaching a clamp to a frame, recites a similar structure discussed above with respect to claim 1, and includes the step of engaging the screw with the clamp by screwing the threaded portion into a threaded engagement with *either* of the first and second through bores such that the head abuts the arm opposite the

threaded engagement and the clearance portion clears the threads of the threaded bore opposite the threaded engagement. (Figs. 4A and 4B; p. 6, lines 3-9.)

Claims 7 and 8, which depend from claim 6, recite the frame as a tubular cycle frame and more particularly that the frame is a tubular bicycle handlebar. (Fig. 2; p. 5, lines 15-18.)

Claim 9 depends from claim 6 and further recites removing the screw from threaded engagement with either of the first and second arms and engaging and tightening the screw in an opposite orientation such that the screw is threadably engaged with the other arm. (p. 6, lines 3-9; Figs. 4A and 4B.)

Independent claim 10 recites a method of manufacturing a symmetrical clamp structure configured as recited in claim 1. Significantly, claim 10 recites forming co-axial cylindrical threaded bores through the distal ends of the first and second arms with each threaded bore having a length less than a select length and forming a clearance portion on the shank of a screw configured to threadably engaging the threaded bores through the distal ends of the first and second arms, the clearance portion being of the select length between the head and the opposite end of the shank. (Fig. 4; p. 6, line 23 – p. 7, line 3.)

Independent claim 11 is directed to a bicycle brake lever comprising a clamp substantially as recited above with regard to claim 1.

Claims 12 and 13 are dependent from claim 11 and further limit the length and the outer diameter of the clearance portion relative to the threaded bores.

Claims 14-17 depend from claim 1, 6, 10 and 11, respectively, and further recite the clearance portion being non-threaded as is illustrated in Figs. 3-4B.

The various embodiments recited in independent claims 6, 10 and 11 and their respective dependent claims provide the advantages discussed above with regard to claim 1.

VI. GROUNDS FOR REJECTION TO BE REVIEWED ON APPEAL

The final Office Action was mailed November 16, 2007. Applicant appeals the following final rejections:

The rejection of claim 10 under 35 USC § 112, second paragraph, as being indefinite for failing to particularly point out and distinctly claim the subject matter which Applicant regards as the invention.

The rejection of claims 1, 4, 6, 7, 8-10 and 14-16 under 35 U.S.C. § 102(b) as being anticipated by Nielsen, U.S. Patent No. 6,186,027.

The rejection of claims 2 and 3 under 35 U.S.C. § 103(a) as being unpatentable over Nielsen, U.S. Patent No. 6,186,027, in view of Steinbock, U.S. Patent No. 6,381,827.

The rejection of claims 2 and 3 under 35 U.S.C. § 103(a) as being unpatentable over Gelbein, U.S. Patent No. 5,584,210, in view of Nielsen, U.S. Patent No. 6,186,027, and further in view of Steinbock, U.S. Patent No. 6,381,827.

The rejection of claims 1, 4, 5, 11 and 17 under 35 U.S.C. § 103(a) as being unpatentable over Gelbein, U.S. Patent No. 5,584,210, in view of Nielsen, U.S. Patent No. 6,186,027.

The rejection of claims 12 and 13 under 35 U.S.C. § 103(a) as being unpatentable over Gelbein, U.S. Patent No. 5,584,210, in view of Nielsen, U.S. Patent No. 6,186,027, and further in view of Steinbock, U.S. Patent No. 6,381,827.

VII. ARGUMENT

A. Summary of the Argument

None of the references cited by the Examiner, Nielsen, U.S. Patent No. 6,186,027, Steinbock, U.S. Patent No. 6,381,827, or Gelbein, U.S. Patent No. 5,584,210, teach a clamp, method for attaching a clamp to a frame, a method of manufacturing a symmetrical clamp structure or a bicycle brake lever as recited in claims 1, 6, 10 and 11, respectively. In particular, none of these references alone nor in combination teach a clamp or method of making a clamp wherein the clamp has a first arm defining a first threaded through bore at its distal end and a second arm having a second threaded through bore at its distal end, with the first and second threaded through bores being essentially coaxial and a screw comprising a head and a shank, with the shank having a threaded portion opposite the head and a clearance portion between the head and the threaded portion with the screw configured so that with a threaded engagement between the threaded portion of the shank and *either* of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other of the first and second arms opposite the threaded engagement, the clearance portion resides in the other of the first and second threaded through bores. In other words, no combination of the references teaches a clamp structure where a screw as described above can be selectively inserted in *either* of the first and second through bores and still perform a clamping function.

B. The Rejection of Claim 10 under 35 U.S.C. § 112

1. Statement of the Relevant Law Pertaining to 35 U.S.C. § 112

35 U.S.C. § 112 provides:

"The specification shall conclude with one or more claims particularly pointing out and distinctly claiming the subject matter which the applicant regards as the invention."

2. The Rejection of Claims 10 under 35 U.S.C. § 112 Is Improper

As best understood, the Examiner maintains the limitation "each threaded through bore having a length less than a select length" renders claim 10 indefinite because the specification does not define "a select length" such that one of ordinary skill in the art would be able to ascertain the scope of the claim. Contrary the position of the Examiner, claim 1 itself makes clear what is meant by a "select length." The term "select length" is used to identify the relationship between the length of the threaded through bores and the clearance portion. Claim 10 recites that each threaded bore has "a length less than a select length" and further subsequently recites having a clearance portion on the shank of a screw "of the select length." Thus, one of skill in the art would immediately recognize claim 1 is reciting a screw having a clearance portion of a select length that is greater than a length of the threaded through bore. This feature is clearly illustrated in Figs. 4A and 4B and one of skill in the art readily would understand that having a clearance portion of a select length greater than the length of the threaded through bore enables the clamp arms 14, 16 to be drawn together as the screw is tightened when deployed as illustrated in Figs. 4A and 4B. Accordingly, Applicant respectfully submits rejection of claim 10 under 35 USC § 112 is improper.

C. The Rejection of Claims 1, 4, 6, 8-10, 14-16 under 35 U.S.C. § 102(b)

1. Statement of the Relevant Law Pertaining to 35 U.S.C. § 102(b)

35 U.S.C. § 102(b) provides:

"A person shall be entitled to a patent unless – (b) the invention was patented or described in a printed publication in this or a foreign country or in public use or on sale in this country, more than one year prior to the date of application for patent in the United States."

To anticipate a claim, a 102(b) reference must teach every element of the claim. "The claim is anticipated only if each and every element as set forth in the claim is found, either expressly or inherently described, in a single prior art reference." *Verdegaal Bros. v. Union Oil Co. of California*, 814 F.2d 628, 631, 2 USPQ2d 1051, 1053 (Fed. Cir. 1987). Anticipation requires the presence in a single prior art reference disclosure of every element of the claimed invention. See, e.g., *Great Northern Corp. v. Davis Core & Pad Co.*, 782 F.2d 159, 165, 228 U.S.P.Q. (BNA) 356, 358 (Fed. Cir. 1986); *Lindemann Maschinenfabrik v. American Hoist and Derrick*, 730 F.2d 1452, 221 U.S.P.Q. (BNA) 481 (Fed. Cir. 1984).

2. The Rejection of Claims 1, 4, 6, 7, 8-10 and 14-16 under 35 U.S.C. § 102(b) Is Improper

Claim 1, and its dependent claims, claims 4 and 14

Claim 1 requires a clamp having a first arm defining a first threaded through bore and a second arm defining a second threaded through bore, with the first threaded through bore and the second threaded through bore being essentially coaxial. Claim 1 further requires a screw having a head and a shank with the shank having a threaded portion opposite the head and a clearance portion between the threaded portion and the head. The screw is configured so that with a threaded engagement between the threaded portion of the shank and *either* of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other of the first and second arms opposite the threaded engagement, the clearance portion resides within the other of the first and second threaded through bores.

The Examiner appears to rely primarily on Fig. 3 of Nielsen in formulating his § 102(b) rejection. While Fig. 3 clearly shows first and second arms 42A, 42B with axially aligned through bores, only the through bore in the second arm 42B is shown as threaded. The Examiner relies on language at column 3, lines 25-28 to support the first through bore in the first arm 42A as being threaded. Nielsen reads at column 3, lines 25-28 as follows:

> "As shown in FIG. 3, the hole in lug 42B is threaded to mate with the threaded shank of screw 40. The hole in lug 42A may but need not be threaded, but is sized so that screw 40 can be rotated therein."

The bolt 40 illustrated in Fig. 3 is a conventional bolt where threads are formed in a threaded portion having an outer diameter equal to an outer diameter of a non-threaded portion

of the shaft. This non-threaded portion of the shaft is adjacent the head of the bolt 40 depicted in Fig. 3. In order for the bolt 40 to be fully received in the axially aligned hole in the first arm 42A, this hole must have an inner diameter greater than the outer diameter of the threaded portion and the non-threaded portion of the bolt 40. Accordingly, if, as suggested in the specification, the hole 42A is threaded, the inner diameter of the threads would have to clear the non-threaded portion of the bolt if the structure is to function as a clamp configuration indicated in Fig. 3. However, in such a configuration the threaded portion of the bolt would necessarily have to clear the threads in 42A without threaded engagement. (This is because as discussed above, the non-threaded portion and the threaded portion of bolt 40 have the same outer diameter.) Thus, the structure taught in Nielsen would not function as a clamp if the bolt 40 were inserted through the second threaded through bore 42B for at least two reasons. First, the threads of the bolt have an outer diameter less than the inner diameter of the threads in the hole 42A and there would thus not be threaded engagement between the threaded portion of the bolt 40 and the threads of 42A. Second, the non-threaded portion of the bolt 40 would interfere with the threads of the second threaded through bore in 42B such that the bolt could only be screwed into threaded through bore 42B up to the point of the non-threaded portion. Accordingly, modification of Fig. 3 as suggested by the Examiner in light of the specification would not meet the limitations of claim 1. Specifically, Nielsen fails to teach a screw configured so that with a threaded engagement between the threaded portion of the shank and *either* of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other of the first and second arms opposite the threaded engagement, the threaded portion resides within the other of the first and second threaded through bores. In other words, while the limitations can be met in part by insertion of the bolt 40 to a threaded hole in 42A as depicted in Fig. 3, it would not be insertable in the second threaded through bore of the second arm 42B in a manner meeting the limitations of claim 1.

It should further be noted that if the hole in 42A had threads of the same inner diameter and pitch as the threads in the hole in the second arm 42B, these limitations would also not be met. This is because, as described above with regard to insertion in the threaded bore 42B, the non-threaded portion of the bolt 40 would interfere with such threads 42A and prevent the head of the bolt 40 from abutting the arm 42A or the arm 42B.

Accordingly, for at least these reasons, Nielsen does not anticipate claim 1.

Claims 4 and 14, which are dependent from claim 1, are not anticipated by Nielsen for the same reasons set forth above with regard to claim 1.

Claim 6, and its dependent claims, claims 8, 9 and 15

Claim 6 is not anticipated by Nielsen for essentially the same reasons set forth above with regard to claim 1. The critical limitation in claim 6 is as follows:

> "engaging the screw with the clamp by screwing the threaded portion into a threaded engagement with *either* of the first and second bores such that the head abuts the arm opposite the threaded engagement and the clearance portion clears the threads of the threaded bore opposite the threaded engagement" (emphasis added)

As discussed above with regard to claim 1, Nielsen does not teach, expressly or inherently, a structure that would allow the screw to be in threaded engagement with *either* of the first and second bores such that the head abuts the arm opposite the threaded engagement and the clearance portion clears the threads of the threaded bore opposite the threaded engagement. In short, the non-threaded portion of the bolt 40 could not be threaded into threaded bore 42B so that these limitations of claim 6 would be met.

Claims 8 and 15 are dependent from claim 6 and are not anticipated for at least the reasons set forth above with regard to claim 6.

Claim 9 is dependent from claim 6 and further recites removing the screw from threaded engagement with *either* of the first and second arms and engaging and tightening the screw in an opposite orientation such that the screw is threadably engaged with the other arm. Claim 9 is thus not anticipated for the same reasons set forth above with regard to claim 6, and moreover because as discussed above with respect to Nielsen, the screw 40 could not be removed after insertion in the first threaded hole of arm 42A and threaded engagement in the second threaded hole of 42B and then inserted in the hole of 42B for threaded engagement with the hole of 42A.

Claim 10 and 16

Independent claim 10, which is directed to a method of manufacturing a symmetrical clamp structure, is not anticipated by Nielsen for the same reasons set forth above with regard to claim 1. Specifically, claim 10 includes the limitation of assembling the clamp by threadably engaging the screw with *either* of the first and second threaded bores such that the head abuts the

arm opposite the threaded engagement and the clearance portion clears the threads of the threaded bore opposite the threaded engagement.

Claim 10 also requires forming co-axial cylindrical threaded bores through the distal ends of the first and second arms, with "each threaded bore having a length less than a select length." Claim 10 further recites forming a clearance portion on the shank "of the select length." In other words, claim 10 requires forming a clearance portion of a length greater than a length of the threaded bores. Referring to Fig. 3 of Neilsen, a non-threaded portion of the bolt 40, which the Examiner argues is a "clearance portion," is depicted with a length less than the length of the bores in 42A or 42B. Thus, Nielsen fails to teach this limitation as well.

Claim 16, which is dependent from claim 10, is not anticipated for the same reasons set forth above with regard to claim 10.

D. <u>The Rejection of Claims 5-7, 20 and 22-23 under 35 U.S.C. § 103(a)</u>

1. Statement of the Relevant Law Pertaining to 35 U.S.C. § 103(a)

The proper standard for rejection of claims under 35 U.S.C. § 103(a) is whether the differences between the claimed subject matter and the prior art are such that the claimed subject matter would have been obvious to one of ordinary skill in the art at the time the invention was made. In *KSR International Co. v. Teleflex Inc.*, 127 S.Ct. 1727, 82 U.P.S.Q.2d (BNA) 1385 (2007), the United States Supreme Court confirmed the following basic obviousness analysis:

In evaluating whether or not an invention is obvious, inquiry into the following three factors must be made:

1. The scope and content of the prior art;
2. The level of ordinary skill in the prior art; and
3. The differences between the claimed invention and the prior art.

See *Graham v. John Deere Co.*, 383 U.S. 1; 86 S. Ct. 684; 15 L. Ed. 2d 545; 148 U.S.P.Q. (BNA) 459 (1966).

The Examiner bears the burden of presenting an unrebutted *prima facie* case of obviousness in order to reject claims under 35 U.S.C. § 103(a). See *In re Deuel*, 51 F.3d 1552, 1557; 34 U.S.P.Q.2d (BNA) 1210 51 F.3d 1552 (Fed. Cir.1995). Thus an applicant on appeal to the Board may overcome the 35 U.S.C. § 103(a) rejection by showing that the Examiner provided insufficient evidence of *prima facie* obviousness.

The Supreme Court confirmed in the *KSR* opinion that a patent examiner or court must articulate a rationale for combining *known elements* from the prior art to formulate an obviousness rejection. The Supreme Court states,

> *"Often, it will be necessary for a court to look to interrelated teachings of multiple patents; the effects of demands known to the design community or present in the marketplace; and the background knowledge possessed by a person having ordinary skill in the art, all in order to determine whether there was an apparent reason to combine the known elements in the fashion claimed by the patent at issue. To facilitate review, this analysis should be made explicit. See* *In re Kahn, 441 F. 3d 977, 988 (CA Fed. 2006)"* *KSR* at 127 S.Ct. 1740-1741.

Thus, though the Supreme Court has jettisoned the "teaching, suggestion, motivation" rubric, an examiner must still identify a viable reason why a person of ordinary skill would have been led to modify the teachings of a reference to arrive at the Applicant's claimed invention. *Ex parte Penhasi*, BPAI Appeal No. 2007-2534 (December 13, 2008).

2. Summary of Argument Concerning 35 U.S.C. § 103(a)

While the Examiner has been able to select references showing in isolation various elements of the claimed combinations in the various claims, the Examiner has not articulated and indeed cannot articulate a viable reason why a person of ordinary skill in the art would have been lead to combine the teachings of the cited references in a manner resulting in Applicant's claimed invention. Moreover, the Examiner has not shown each of the recited elements in the prior art.

3. The Rejection of Claims 2 and 3 under 35 U.S.C. § 103(a) Is Improper

Claims 2 and 3 are dependent from claim 1. Claim 2 recites the clearance portion of claim 1 has an has an outer diameter sized to clear the first and second threaded bores and a length at least equal to the axial length of each threaded bore. Claim 3 recites a length of the clearance portion exceeds an axial length of each threaded bore.

As discussed above with regard to the rejection of claim 1 under 35 USC § 102(b), Nielsen fails to teach a clamp having axially aligned threaded through bores and a screw configured so that with a threaded engagement between the threaded portion of the shank and *either* of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other arm, the clearance portion resides within the other

of the first and second threaded through bores. Steinbock does not provide a teaching of these missing elements. Steinbock, which is directed to a method for maintaining a clamping force between bolted parts in high temperatures, uses an elongated fastener 20 with threaded portions 22, 24 at opposite ends of the shank separated by a reduced diameter portion indicated at reference 23. As best understood, the significance of the reduced diameter portion is not the function it performs itself, which is that the reduced diameter portion can fit within a smaller diameter orifice, but that a larger diameter threaded portion 24 is provided which allows a large sheer area where the fastener is threadably engaged with the internal threads 26 in the flange 14. Thus, rather than teaching the desirability of a reduced diameter portion, Steinbock is teaching the advantage of increasing the diameter of a threaded portion to increase the sheer area at the point of threaded engagement. The Examiner fails to indicate why this teaching would motivate a person of skill in the art to provide a clamp structure with two opposing threaded bores and screw having a clearance portion between a head and a threaded portion configured so that with a threaded engagement between the threaded portion of the shank and *either* of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other, a clearance portion resides in the other of the first and second threaded through bores. Indeed, referring to Figs. 2-8 of Steinbock, the purported reduced diameter portion of the shaft 23 does not reside within a threaded bore. In summary, the Examiner has not identified and cannot articulate a reason why one skilled in the art would have modified the teachings of Nielsen in view of Steinbock to yield the invention recited in claim 1.

Claim 2 requires the clearance portion have an outer diameter sized to clear the first and second threaded bores and a length at least equal to the axial length of each threaded bore. Neither of these elements is taught by Nielsen. The "clearance portion" 23 of Steinbock does not appear to be intended to have a length greater than the internal threads 26 illustrated in Figs. 2-7. In any event, while this teaching may arguably be derived from Fig. 8, there is no articulated reason why one of skill in the art would have provided a reduced diameter to the non-threaded portion of the bolt 40 of Nielsen and locate such a clearance portion within the first or second threaded through bores. At page 7 of the final Office Action, the Examiner's statement that "Steinbock teaches that the length of the clearance portion (20) exceeds an axial length of each threaded bore for the purpose of allowing a large sheer area which can prevent stripping of the threads", citing column 5, lines 35-42 of Steinbock, contained at page 7, simply is not true. As

discussed above, the cited language pertains to the advantage of having an enlarged threaded portion 24 which provides a large sheer area. Taking the Examiner's argument to its logical extreme, the sheer area could be increased by having the reduced diameter portion extend to only a single thread on the shaft to purportedly maximize the sheer area, but common sense of course dictates that it would in fact minimize the sheer area and lead to failure of the clamp taught in Steinbock.

Reference to Gelbein at page 7 of the final Office Action does not remedy the deficiencies of Steinbock and Nielsen, and in fact the Examiner does not even so argue.

4. The Rejection of Claims 1, 4, 5, 11 and 17 under 35 U.S.C. § 103(a) Is Improper

The Examiner's statement of the purported teaching of Gelbein, U.S. Patent No. 5,584,210, is set forth at the eighth page of the final Office Action. Applicant does not contest the Examiner's position with regard to what Gelbein teaches and what Gelbein fails to disclose. In particular, as admitted by the Examiner, Applicant notes that Gelbein does not disclose coaxial first and second threaded through bores or a screw having a clearance portion between a threaded portion and a head and the screw being configured so that with a threaded engagement between the threaded portion of the shank and *either* of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other of the first and second arms opposite the threaded engagement, the clearance portion resides in the other of the first and second threaded through bores.

The Examiner contends that these elements are taught by Nielsen. However, as discussed above with respect to rejection of claim 1 under 35 USC § 102(b), Nielsen does not teach these elements. In essence, there is no teaching that the bolt 40 of Nielsen can be threadably engaged with *either* the threaded bore in 42A or 42B with clearance portion residing in the other threaded portion.

Moreover, the Examiner fails to articulate a reason why or even how one of ordinary skill in the art could modify the device of Gelbein to provide a second threaded through bore which is essentially coaxial with the first threaded through bore. As characterized by the Examiner, this would require, for example, first arm (right side of element 32) being broken off and moved along an axis corresponding to the hole 56 of the second arm (the left side of 32). This would

defy common sense and would allow only one of the "arms" to engage a handle bar or other frame portion and would thus defeat the purpose of providing the two coaxial clamping elements shown in Gelbein.

Regarding claim 4, in the context of the claims, a "proximal" end means an end attached to body and a "distal" end means an end that is free from the body. Referring to Fig. 3 of Gelbein, the axial bores appear to be on the proximal end of what is characterized as the first and second arms of Gelbein. Thus, Gelbein would not teach the first and second threaded bores being at the distal end of the first and second arms of the clamp as required in claim 1. Accordingly, the elements of claim 4 are not taught by any combination of Gelbein and Nielsen.

Claim 5 is not obvious for the same reasons discussed above with respect to claims 1 and 4.

Regarding independent claim 11, it requires, like claim 1, first and second arms with the first arm having a first threaded through bore and a second arm having a second threaded through bore with the first and second threaded through bores being essentially coaxial. As discussed above with regard to claims 1 and 4, the structure of Gelbein would have to be severely modified in order to align the bores of the first and second arms (as characterized by the Examiner). Besides the Examiner failing to articulate why one of skill in the art would modify Gelbein to align the arms in this manner in view of Nielsen, one skilled in the art would actually be led away from reconfiguring Gelbein in this manner because such a modification of Gelbein would severely diminish the effectiveness of the clamps of Gelbein.

Regarding claim 17, which is dependent from claim 11, claim 17 would not be obvious for the same reasons set forth above with regard to claim 11.

Claims 12 and 13 stand rejected over Gelbein in view of Nielsen and in further view of Steinbock. As an initial matter, Applicants have set forth above why Gelbein in view of Nielsen fails to teach or suggest the combination of elements recited in independent claim 11, from which claims 12 and 13 depend. Furthermore, as discussed above with respect to claims 2 and 3, Steinbock does not provide any teaching that the length of the clearance portion 20 should exceed an axial length of *each* threaded through bore. The Examiner is simply wrong that Steinbock teaches at column 5, lines 35-42 providing a clearance portion 20 with a length that exceeds an axial length of each threaded bore allows for "a large sheer area which can prevent stripping of the threads (column 5, lines 35-42)". As pointed out above, what Steinbock is really

teaching is providing an increased outer diameter of a threaded portion provides a large sheer area and Steinbock in no way relates the length of the clearance portion to a large sheer area provided by threads.

VIII. CLAIMS APPENDIX

1. A clamp comprising:

a first arm having a distal end defining a first threaded through bore;

a second arm having a distal end defining a second threaded through bore, wherein the first threaded through bore and the second threaded through bore are essentially coaxial; and

a screw comprising a head and a shank, the head being at one end of the shank and the shank having a threaded portion at a second end opposite the first end and a clearance portion between the threaded portion and the head, the screw being configured so that with a threaded engagement between the threaded portion of the shank and either of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other of the first and second arms opposite the threaded engagement, the clearance portion resides within the other of the first and second threaded through bores.

2. The symmetric clamp structure of claim 1 wherein the clearance portion has an outer diameter sized to clear the first and second threaded bores and a length at least equal to the axial length of each threaded bore.

3. The symmetric clamp structure of claim 1 wherein a length of the clearance portion exceeds an axial length of each threaded bore.

4. The symmetric clamp structure of claim 1 wherein each of the first and second arms have a proximal end attached to a bicycle component.

5. The symmetric clamp structure of claim 4 wherein the bicycle component is a brake lever.

6. A method of attaching a clamp to a frame comprising:

providing a frame;

providing a symmetric clamp structure comprising a first arm having a distal end defining a first threaded bore, a second arm having a distal end defining a second threaded bore wherein the first threaded bore and the second threaded bore are essentially coaxial and have essentially the same size and pitch threading;

providing a screw comprising a head and a shank, the head being at one end of the shank and a threaded portion being at a second end of the shank opposite the first end, the threaded portion being sized to threadably engage both the first and second threaded bores, the shank further comprising a clearance portion between the threaded portion and the head;

engaging the screw with the clamp by screwing the threaded portion into a threaded engagement with either of the first and second bores such that the head abuts the arm opposite the threaded engagement and the clearance portion clears the threads of the threaded bore opposite the threaded engagement;

placing the clamp over the frame so that the frame is received between the first and second arms of the clamp; and

tightening the screw thereby driving the distal ends of the first and second arms toward each other, thereby attaching the clamp to the frame.

7. The method of claim 6 wherein the frame is a tubular bicycle frame.

8. The method of claim 6 wherein the frame is a tubular bicycle handlebar.

9. The method of claim 6 further comprising removing the screw from threaded engagement with either of the first and second arms and engaging and tightening the screw in an opposite orientation such that the screw is threadably engaged with the other arm.

10. A method of manufacturing a symmetrical clamp structure comprising:

providing a clamp body having a first arm having a distal end and a second arm having a distal end with the distal end of the first arm and the distal end of the second arm being substantially adjacent to each other and defining a gap between the arms;

forming co-axial cylindrical threaded bores through the distal ends of the first and second arms, each threaded bore having a length less than a select length;

providing a screw having a head at one end and a threaded shank extending from the head to an opposite end with the threaded shank being sized to threadably engage the threaded bores through the distal ends of the first and second arms;

forming a clearance portion on the shank of the select length between the head and the opposite end of the shank such that the clearance portion extends toward but not to the opposite end, leaving a portion of the shank opposite the head threaded;

assembling the clamp by threadably engaging the screw with either of the first and second threaded bores such that the head abuts the arm opposite the threaded engagement and the clearance portion clears the threads of the threaded bore opposite the threaded engagement.

11. A bicycle brake lever comprising:

a housing;

a lever pivotably attached to the housing;

a clamp attached to the housing, the clamp comprising:

first and second arms configured to receive a bicycle handlebar axially therebetween, each of the first and second arms having a distal end, the distal ends having a space therebetween, the first arm further having a first threaded through bore at its distal end and the second arm further having a second threaded through bore at its distal end, the first and second threaded through bores being essentially coaxial; and

a screw comprising a head and a shank, the head being at one end of the shank and the shank having a threaded portion at a second end opposite the first end and a clearance portion between the threaded portion and the head, the screw being configured so that with a threaded engagement between the threaded portion of the shank and either of the first threaded through bore of the first arm or the second threaded through bore of the second arm and the head abutting the other of the first or second arms opposite the threaded engagement, the clearance portion

resides within the other of the first and second threaded through bores, such that there is no threaded engagement between the threaded portion of the shank and the other of the first and second threaded through bores.

12. The bicycle brake lever of claim 11 wherein the clearance portion has an outer diameter sized to clear the first and second threaded bores and a length at least equal to the axial length of each threaded bore.

13. The bicycle brake lever of claim 11 wherein a length of the clearance portion exceeds an axial length of each threaded bore.

14. The symmetric clamp structure of claim 1 further comprising the clearance portion being non-threaded.

15. The method of claim 6 further comprising the clearance portion being non-threaded.

16. The method of claim 10 further comprising the clearance portion being non-threaded.

17. The bicycle brake of claim 11 further comprising the clearance portion being non-threaded.

IX. EVIDENCE APPENDIX

Enclosed please find copies of the following references relied upon by the Examiner as to the grounds of rejection to be reviewed upon appeal:

1. Nielsen, U.S. Patent No. 6,186,027.
2. Steinbock, U.S. Patent No. 6,381,827.
3. Gelbein, U.S. Patent No. 5,584,210.

X. RELATED PROCEEDINGS APPENDIX

None

XI. CLOSING REMARKS

For the foregoing reasons, Applicant submits that the rejection of claim 10 under 35 U.S.C. § 112, second paragraph is improper, that the rejection of claims 1, 4, 6, 7, 8-10 and 4-16 under 35 U.S.C. § 102(b) is improper, that the rejection of claims 2-3 under 35 U.S.C. § 103(a) is improper, that the rejection of claims 1, 4, 5, 11 and 17 under 35 USC § 103(a) is improper and that rejection of claims 12 and 13 under 35 U.S.C. § 103(a) is improper. Thus, Applicant respectfully submits that claims 1-17 are therefore patentable. Accordingly, Applicant respectfully requests that the rejections of the Examiner be reversed.

The undersigned hereby authorizes the charge of any required fees not included or any deficiency of fees submitted herewith to be charged to deposit account number 19-5117.

Respectfully submitted,

Date: 5/15/08

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IX. EVIDENCE APPENDIX

1. Nielsen, U.S. Patent No. 6,186,027.
2. Steinbock, U.S. Patent No. 6,381,827.
3. Gelbein, U.S. Patent No. 5,584,210.

US006186027B1

(12) **United States Patent**
Nielsen

(10) **Patent No.:** **US 6,186,027 B1**
(45) **Date of Patent:** **Feb. 13, 2001**

(54) **HANDLEBAR STEM ASSEMBLY FOR BICYCLE FORK**

(76) Inventor: **Peter M. Nielsen**, 21 Nut Island Ave., Quincy, MA (US) 02169

(*) Notice: Under 35 U.S.C. 154(b), the term of this patent shall be extended for 0 days.

(21) Appl. No.: **09/511,055**

(22) Filed: **Feb. 23, 2000**

(51) **Int. Cl.**[7] **B62K 21/18**
(52) **U.S. Cl.** **74/551.1**; 403/365
(58) **Field of Search** 74/551.1, 551.3; 280/279, 280; 403/365, 299

(56) **References Cited**

U.S. PATENT DOCUMENTS

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5,553,511 * 9/1996 Marui 74/551.1
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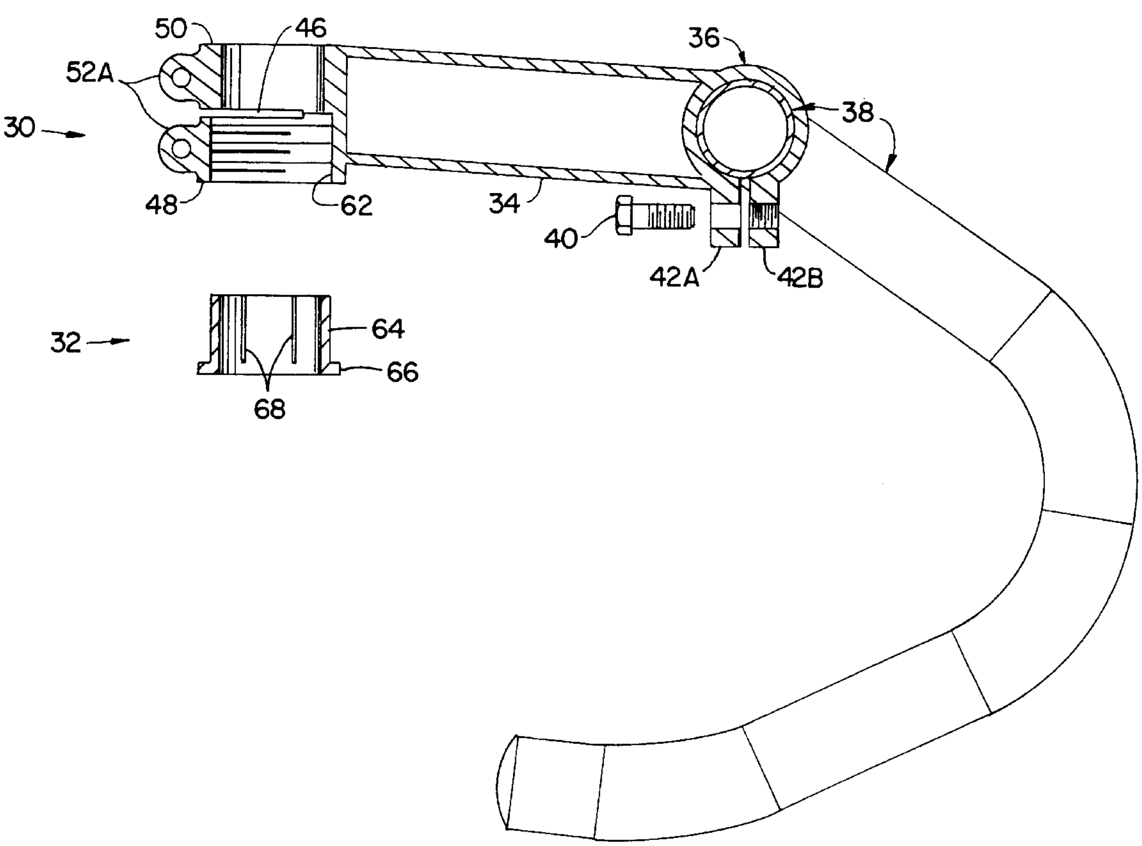
Primary Examiner—Mary Ann Green
(74) *Attorney, Agent, or Firm*—Pandiscio & Pandiscio

(57) **ABSTRACT**

A bicycle stem assembly comprising a bicycle stem having a novel ways for securing it to an unthreaded bicycle fork steerer tube, the novel means featuring a castellated bushing for pre-loading the upper bearing for the steerer tube and a clamp carried by the stem for selectively locking the stem and bushing to the steerer tube.

19 Claims, 5 Drawing Sheets



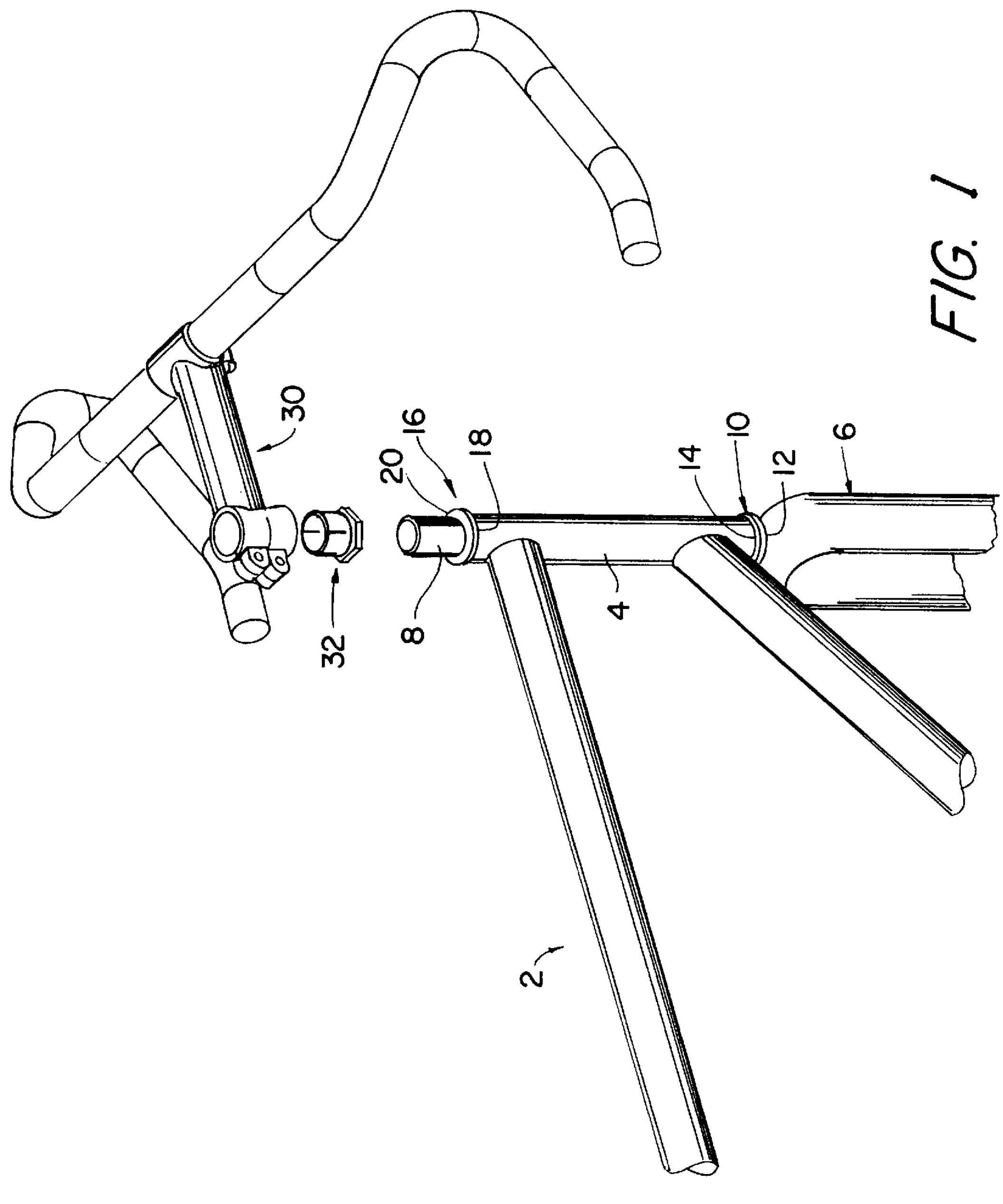


FIG. 1

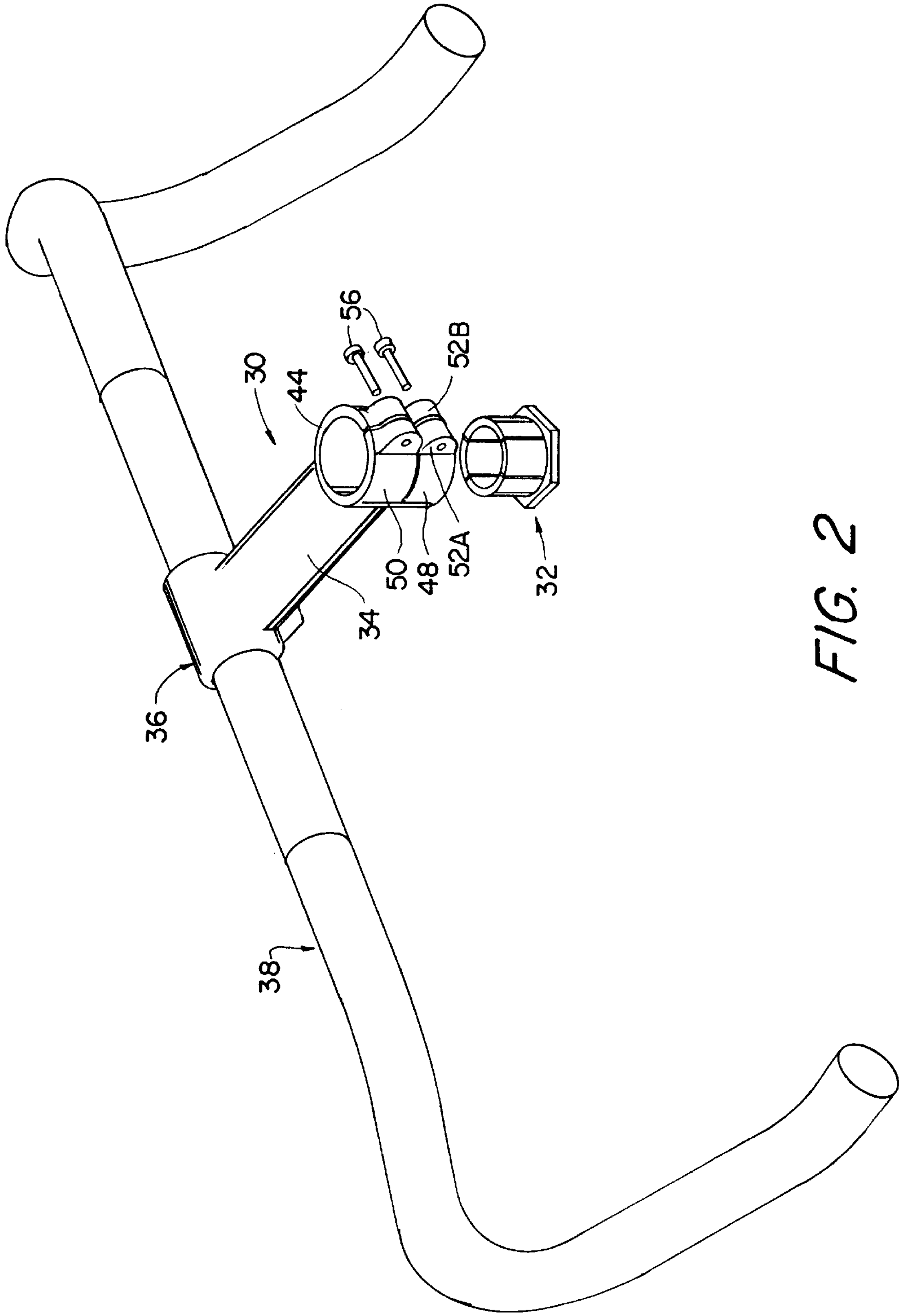


FIG. 2

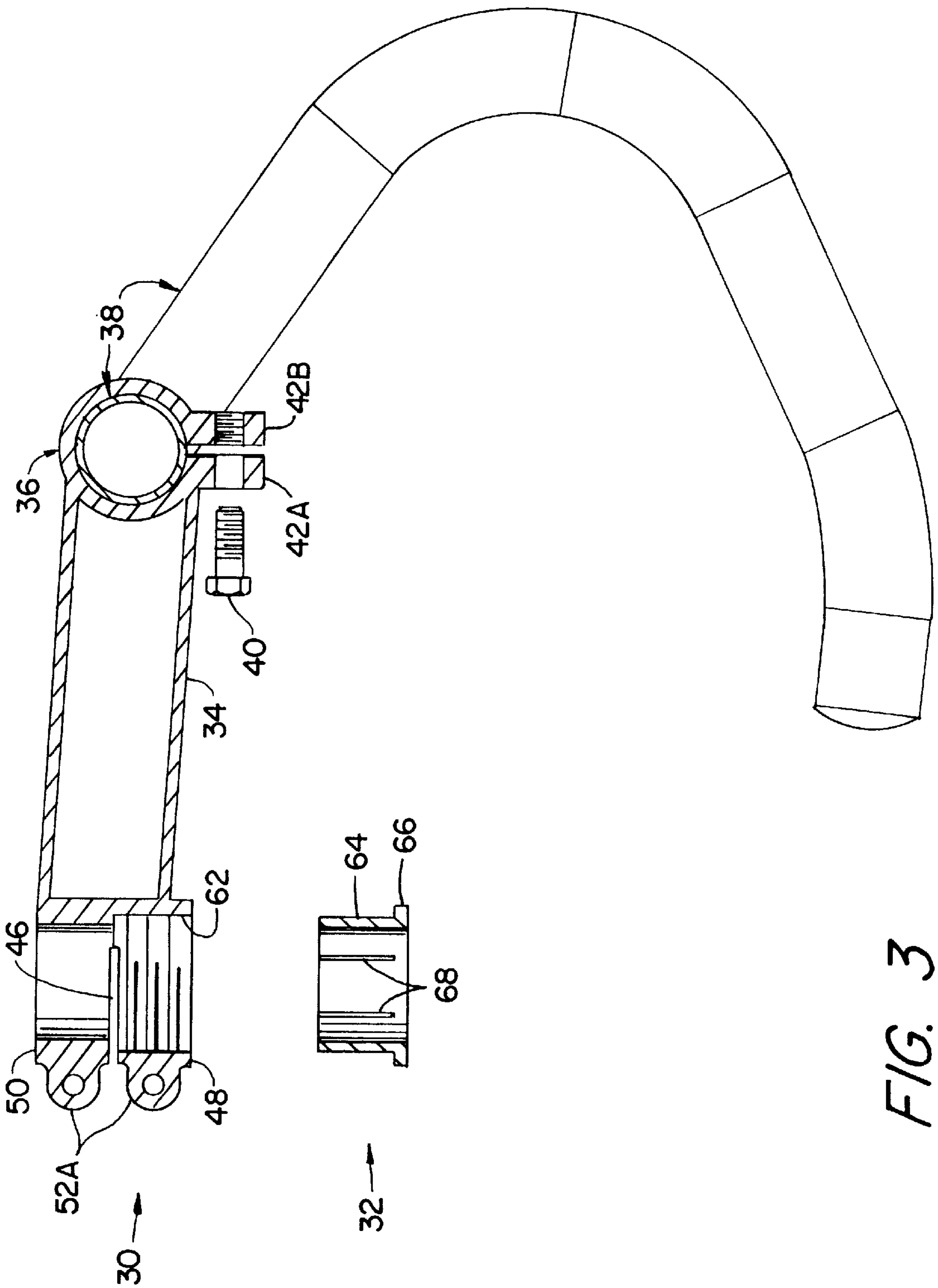


FIG. 3

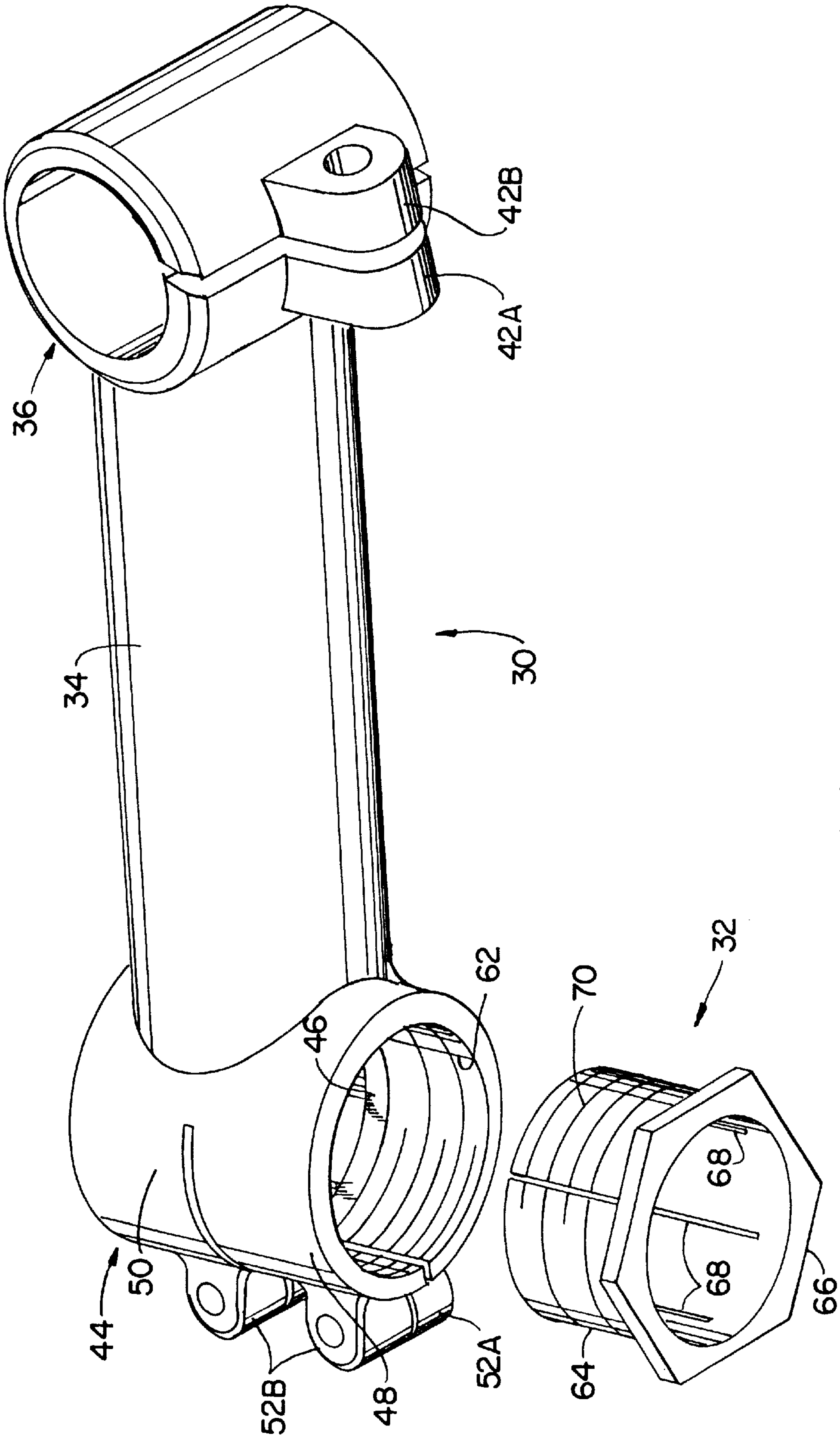


FIG. 4

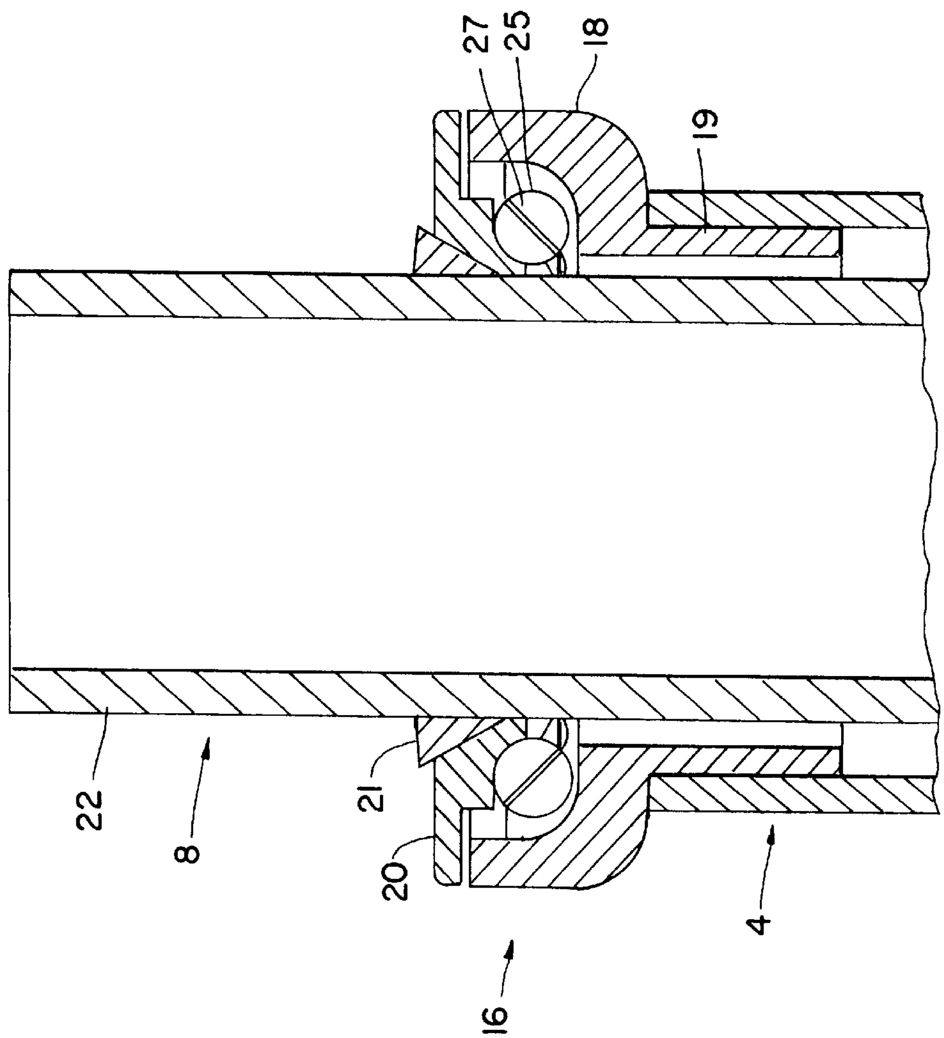


FIG. 5

HANDLEBAR STEM ASSEMBLY FOR BICYCLE FORK

BACKGROUND OF THE INVENTION

This invention relates to improvements in bicycle handlebar stems and more particularly to an improved assembly for securing a bicycle stem to a bicycle fork steerer tube.

DESCRIPTION OF THE PRIOR ART

The handlebar of a bicycle is coupled to the bicycle frame by means of a stem having a vertical tube which is secured to the steerer tube of the fork to which the front wheel is attached. Two different approaches have been used extensively for coupling the bicycle handlebar stem to the steerer tube. One approach involves division of an external screw thread on the upper end of the steerer tube onto which is secured the top race of the upper bearing headset assembly, and a lock nut, with the top end of the steering tube terminating flush with or just below the lock nut. With this approach, the vertical tube of the stem is inserted into the steerer tube and carries a threaded fixing stud which in turn carries a wedge nut at its lower end. The fixing stud has a head that is accessible through the top end of the vertical tube of the stem. When the stud is turned, the wedge nut is pulled upwardly against a compensating tapered surface at the lower end of the vertical tube, causing the wedge to be displaced laterally and to tighten against the inner surface of the steerer tube, whereby the stem acts as an extension of the steerer tube.

The second approach uses a stem that comprises a dual clamp section that surrounds the steerer tube. In this arrangement, the length of the steerer tube is extended so as to accommodate the clamp section of the stem. The upper race of the upper bearing is not screwed to the steerer tube. In order to permit pre-loading the steering headset bearing, a star nut is located inside the steerer tube, and a preload cap is engaged with the upper end of the steerer tube, and a threaded studs extends through the cap and is screwed into the star nut. Turning the threaded stud varies the force with which the cap forces the stem into engagement with the outer race of the upper bearing headset and thereby determines the pre-loading of the bearing.

The first approach is complicated and relatively expensive. The second approach offers the advantage of using a lighter weight stem. However, the second approach suffers from the limitation that it is easy to make a mistake in properly positioning the star nut in the steerer tube, and correcting the mistake is difficult since the star nut, by design, tightly grips the steerer tube. Damage caused by forcing removal of the star nut is especially disadvantageous in the case of expensive light weight bicycle frames. As an alternative to the star nut, a small device which expands radially and has internal threads is inserted into the steerer tube in place of the star nut. The expandable device is more easily removable than the star nut, but it suffers from the disadvantage that it makes a tenuous grip on the steerer tube.

A further problem with the foregoing arrangements is that they make it difficult, if not impossible, to pass a brake cable through and out of the upper end of the steerer tube, thus preventing use of the brake assembly invention disclosed in my U.S. Pat. No. 5,803, 207, issued Sep. 8, 1998.

OBJECTS AND SUMMARY OF THE INVENTION

The primary object of the invention is to provide a new assembly for positively and securely locking a bicycle stem to an unthreaded steerer tube.

Another object is to provide a novel stem-securing means which is simple and inexpensive to manufacture and use.

Still another object is to provide a bicycle handlebar stem having novel means for securing it to a steerer tube.

A further object is to provide a novel bicycle stem/steerer tube combination that facilitates pre-loading of the steerer tube bearings.

Another object is to provide a method of attaching a bicycle stem to a steerer tube which facilitates pre-loading of the upper bearing for the steerer tube.

A more specific object is to provide an arrangement for securing a stem to a bicycle fork steerer tube that allows a brake cable to extend from brake actuating means carried by the fork up through and out of the upper end of the steerer tube.

The foregoing objects are achieved by utilizing with a non-threaded steerer tube a castellated externally-threaded locking bushing that fits onto the upper end of the steerer tube in position to engage the outer race of the upper steering head bearing, and a bicycle stem that comprises a first split clamp section that closely surrounds the tube and is adapted to be compressed into tight engagement with the steerer tube, and a second split clamp section that surrounds and makes a screw thread connection with the castellated bushing and is adapted to be compressed to squeeze the bushing so as to lock it to the steerer tube. The invention includes a method of assembling the stem and bushing with the steerer tube that facilitates pre-loading of the upper bearing associated with the steerer tube. Other features and advantages of the invention are disclosed or rendered obvious by the following detailed specification.

THE DRAWINGS

FIG. **1** is a partially exploded fragmentary isometric view of a bicycle frame and fork incorporating a handlebar stem and a locking bushing embodying the invention;

FIG. **2** is a fragmentary isometric view from a different angle of the same stem and locking bushing:

FIG. **3** is a sectional view of the same stem and locking bushing;

FIG. **4** is an enlarged isometric view of the stem and locking bushing; and

FIG. **5** is a cross-sectional view of a preferred form of upper bearing assembly for coupling the steerer tube to the bicycle head tube.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings, there is shown a portion of a bicycle frame **2** having at its front end an open-ended tubular member **4** (commonly called a "head tube") which rotatably receives a steerer tube **8** which is an extension of a bicycle wheel- supporting fork member **6**. The fork and the tubular member are coupled by a conventional lower steering head bearing assembly **10** which comprises a first (lower) race **12** that is secured to fork **6** (or the lower end of steerer tube **8**), and a second (upper) outer race **14** that is secured to the lower end of tubular member **4**. The races of bearing assembly **10** may be affixed, i.e., secured in place, by means well known to persons skilled in the art, e.g., a press fit or a screw connection. Although not shown, it is to be understood that bearing assembly **10** also includes a plurality of ball bearings in a retainer ring that is disposed between the two races **12** and **14**.

A second upper bearing assembly **16** is located at the upper end of the tubular member **4**. As illustrated in FIG. **5**,

the bearing assembly **16** comprises a first (lower) race **18** having a tubular extension **19** that is secured to head tube **4**. This may be done by way of a press-fit, or by a screw thread connection, or by some other means known to persons skilled in the art. The second (upper) race **20** of bearing assembly **16** surrounds the upper end portion **22** of the steerer tube but is not affixed to that tube. Instead it is free to be rotated relative to steerer until it is locked by action of the locking bushing **32** hereinafter described. The inner surface of upper race **20** is tapered to accommodate a tapered centering ring **21** which is split at one point along its circumference so as to allow it to be compressed radially. Bearing assembly **16** also includes a ball bearing/retainer assembly comprising plurality of ball bearings **25** located in holes in a retainer ring **27** that is disposed between the two races **18** and **20**.

The illustrated apparatus further includes a novel stem **30** and a castellated threaded locking bushing **32**. Referring to FIGS. **1–4**, the stem **30** comprises a body portion **34** having at one end a tubular member **36** for accommodating a typical handlebar **38**. The cylindrical member **36** is split and is provided with a pair of lugs **42**A, **42**B having aligned holes for receiving a screw **40** that is used to draw the lugs together so as to compress the tubular member **36** about the handlebar **38**, thereby locking the handlebar in place. As shown in FIG. **3**, the hole in lug **42**B is threaded to mate with the threaded shank of screw **40**. The hole in lug **42**A may but need not be threaded, but is sized so that screw **40** can be rotated therein.

The other or rear end of the stem body **34** is provided with a second tubular member **44** which is oriented so that its axis extends at a right angle to the axes of stem body **34** and tubular member **36**. The tubular member **44** is slotted transversely to its axis as shown at **46**, so as to form two clamp sections **48** and **50**. Slot **46** extends through an angle of at least **1800** but less than **3600** along the circumference of tubular member **44**. As seen best in FIGS. **2** and **4**, each of these clamp sections in turn is split lengthwise at the "six o'clock" position determined relative to body portion **34**.

Each of the clamp sections **48** and **50** is provided with a pair of lugs **52**A, **52**B extending therefrom adjacent where they are split. The lugs **52**A, **52**B of each clamp section have aligned holes for receiving a screw **56**. Although not shown, the holes in lugs **52**A are threaded to make a screw connection with screws **56**. The holes in lugs **52**B may, but need not, be threaded. By turning screws **56** in a given direction it is possible to draw together the lugs **52**A, **52**B of clamp sections **48** and **50**, thereby effectively reducing the inside diameter of those clamp sections. It should be noted that clamp section **48** differs from clamp section **50** in that its inner surface is provided with a screw thread as represented schematically at **62**. The two clamp sections are made so that when they are not compressed by action of screws **56**, their internal diameters are slightly larger than the outer diameter of the upper end of the steerer tube, preferably about 0.005 inch larger.

The bushing **32** comprises a hollow body section **64** and a peripheral flange **66**. The body section **64** is cylindrical, while the flange **66** preferably has a polygonal edge configuration (e.g., hexagonal) to permit it to be grasped by a suitable wrench. Bushing **32** is castellated in the sense that its hollow tubular body portion **64** is slit longitudinally at several locations as indicated at **68**. The slits **68** commence at or near flange **66** and extend through the opposite end edge of body section **64**, as seen best in FIG. **4**. Additionally, the exterior surface of the body portion of bushing **32** has a screw thread as represented schematically at **70** in FIG. **4**. Bushing **32** is sized so that its exterior screw thread **70** will

make a loose screw connection with the internal screw thread **62** of clamp section **48** when the latter is in its uncompressed condition, i.e., when its screw **56** in lug **52**B is completely, or nearly completely, unscrewed from lug **52**A.

How the handlebar stem and the locking bushing are assembled onto and secured to the steerer tube so as to achieve selective preloading of the upper bearing assembly will now be described. Assume that steerer tube **8** is disposed in head tube **4** as shown in FIG. **1**, with the two races of bearing **10** secured in place as previously described, the lower race **18** of bearing **16** secured to head tube **4**, and the upper race **20** and the centering ring **21** surrounding but not secured to steerer tube **8**. With both of the screws **56** backed off so that the clamp section **48** is expanded enough to accommodate and make a screw connection with bushing **32** and clamp section **50** is large enough to make a close sliding fit on the upper end of the steerer tube, bushing **32** is screwed substantially all the way into clamp section **48**. Then, with the bushing in place, stem **30** is slipped onto the steerer tube **22** so that clamp section **50** surrounds the upper end of the steerer tube and the flange **66** of the bushing lies close to or lightly engages the upper race **20** of the upper bearing assembly **16**. Then the screw **56** of clamp section **50** is turned so as to cause that clamp section to be drawn into tight clamping relation with the upper end **22** of steerer tube **8**. Thereafter, bushing **32** is rotated in a direction to move it away from clamp section **50**. As the bushing is backed out of clamp section **48**, its flange **66** forces centering ring **21** to exert a downward pressure on the upper race **20** of bearing assembly **16**. Centering ring **21** functions to (a) locate the race **20** radially relative to the steerer tube so that the steerer tube is centered relative to that race and (b) also apply an axial preload to bearing **16**. The axial pressure applied to the centering ring compresses it down onto the mating taper of the inner surface of race **20**, and also radially inward to firmly grip the steerer tube, much like a collet functions. The bushing's position is adjusted so that it exerts a suitable load on bearing assembly **16** that is calculated to eliminate or acceptably minimize axial play of the steerer tube in the frame's head tube **4**. Once the desired bearing preloading is achieved, the screw **56** of clamp section **48** is turned in a direction to compress that clamp section radially inward to an extent that compresses the castellated body portion **64** of the bushing into a gripping and locking relations with the steerer tube, with the result that the preload applied to the upper bearing assembly by the bushing will remain fixed.

The foregoing design offers a number of advantages. For one thing, it is simpler than prior arrangements that are designed to permit adjustable preloading of the steerer tube bearing assemblies. Another advantage is that it avoids possible damage to the steerer tube as can occur when using a star nut. It also does not require the steerer tube to be threaded. Additionally, the preloading of the bearing assembly is easily adjusted, requiring in turn only loosening of the screw **56** associated with clamp section **148**, rotation of bushing **32**, and then re-tightening the screw **56**. Another important advantage is that the interior of the steerer tube remains unobstructed, so that it is now possible to pass a brake cable through and out of the upper end of the steerer tube, thereby making it possible to use brake assemblies as disclosed in my U.S. Pat. No. 5,803,207, issued Sep. 8, 1998.

Still another advantage is that it is not necessary to cut the steerer tube to a predetermined length, as is required with prior systems. This invention allows the stem to be moved up or down within limits on the steerer tube, while still

allowing the castellated bushing to provide desired preloading of the bearings. The new stem architecture further offers the advantage of reduced cost because of its simplicity. Of additional benefit is the fact that bearings **10** and **16** are of conventional design. Still other advantages will be obvious to persons skilled in the art from the foregoing description and the drawings.

Obviously the apparatus as shown in the drawings may be modified without departing from the spirit of the invention, for example, the two clamp members **48** and **50** may be formed as separate members and welded to the stem body **34**, rather than constituting portions of a common cylinder. Additionally, the lugs **42**A, **42**B (and also the lugs **52**A, **52**B) may be coupled together by a screw and nut arrangement, thereby eliminating the need to a threaded hole in one or both lugs. Similarly, the tubular member **36** at the front end of the stem body **34** also may be modified without affecting the mode of attachment of the stem to the steerer tube. Also various forms of angular contact bearings that require some axial preload to function properly may be used as the lower and upper bearings **10** and **16**. Still other modifications will be obvious to persons skilled in the art.

What is claimed is:

1. A bicycle fork and handlebar stem assembly comprising:

a fork steerer tube;

a castellated externally-threaded bushing that fits on an end portion of said steerer tube; and

a handlebar stem comprising a body section and first and second split tubular clamp sections formed integral with said body section, said first clamp section surrounding said steerer tube, and said second clamp section surrounding and making a screw thread connection with said bushing, said first clamp section being adapted to be compressed into tight locking engagement with said steerer tube and said second clamp section being adapted to be compressed so as to squeeze said bushing into tight locking engagement with said steerer tube.

2. An assembly according to claim **1** wherein said fork steerer tube is an extension of a bicycle fork for supporting a bicycle wheel, and further wherein said second clamp section is located between said first clamp section and said bicycle fork.

3. An assembly according to claim **1** wherein said bushing comprises a castellated tubular body section and a flange section located at one end of said body section, and further wherein said flange section has an outer diameter greater than the outer diameter of said end portion of said steerer tube.

4. An assembly according to claim **1** wherein each of said first and second clamp sections comprises first and second diametrically opposed clamping portions having first ends thereof formed integral with said body section and second ends having first and second mutually confronting lugs, and screw means connecting said first and second lugs for drawing said first and second clamping portions toward one another whereby to force said first clamp section into tight engagement with said steerer tube and cause said second clamp section to radially compress said bushing into tight engagement with said steerer tube.

5. An assembly according to claim **4** wherein said screw means comprises a threaded bolt that extends through a hole in said first lug and screws into a threaded hole in said second lug.

6. An assembly according to claim **4** wherein one end of said bushing projects out of said second clamp section in position to engage a bearing assembly associated with said steerer tube.

7. An assembly according to claim **1** wherein each of said first and second clamp sections comprises first and second diametrically opposed clamping portions having first ends thereof formed integral with said stem section and second ends having first and second mutually confronting lugs, and screw means connecting said first and second lugs for drawing said first and second clamping portions toward one another whereby to force said first clamp section into tight engagement with said steerer tube and cause said second clamp section to radially compress said bushing into tight engagement with said steerer tube.

8. An assembly according to claim **7** wherein said first and second clamp sections are adjacent to and aligned with one another.

9. An assembly according to claim **8** wherein said first and second clamp sections are formed at one end of said body section.

10. An assembly according to claim **9** wherein said body section has a second end opposite to said first end, and further wherein said second end has means for supporting a handlebar.

11. An assembly according to claim **7** wherein said screw means comprises a threaded bolt that extends through a hole in said first lug and screws into a threaded hole in said second lug.

12. In combination with a bicycle frame having a tubular member having first and second open ends, a wheel-supporting fork having a steerer tube rotatably disposed in and extending through said tubular member with said steerer tube having an outer end that projects from said second open end of said tubular member, a lower bearing assembly comprising a first race secured to said fork and a second race secured to said first open end of said tubular member, and an upper bearing assembly comprising a first race secured to said second open end of said tubular member and a second race surrounding said outer end of said steerer tube;

a castellated bushing closely surrounding said outer end of said steerer tube, said bushing being disposed so that one end thereof confronts said second race of said upper bearing assembly and being formed so as to be capable of axial and rotational movement relative to said steerer tube; and

a handlebar stem comprising a stem section adapted to securely support a handlebar device and first and second split tubular clamp sections formed integral with and extending out from said stem section, said first clamp section closely surrounding said outer end of said steerer tube, and said second clamp section surrounding and making a screw thread connection with said bushing, said first clamp section being adapted to be compressed into tight locking engagement with said outer end of said steerer tube and said second clamp section being adapted to be compressed so as to squeeze said bushing into tight locking engagement with said outer end of said steerer tube.

13. An assembly according to claim **12** wherein each of said first and second clamp sections comprises first and second diametrically opposed clamping portions having first ends thereof formed integral with said stem section and second ends having mutually confronting lugs, and screw means connecting said lugs for drawing said first and second clamping portions toward one another whereby to force said first clamp section into tight engagement with said outer end of said steerer tube and cause said second clamp section to radially compress said bushing into tight engagement with said outer end of said steerer tube.

14. A method of assembling the components of the assembly defined by claim **13** comprising the steps of:

(a) screwing said bushing into said second clamp section;

(b) positioning said stem so that said first and second clamp sections surround said outer end of said steerer tube;

(c) manipulating said screw means of said first clamp section so as to lock said first clamp section to said outer end of said steerer tube;

(d) rotating said bushing so as to cause it to engage and force said second race of said upper bearing assembly toward said first race of said upper bearing assembly, so as to preload said bearing assembly; and

(e) manipulating said screw means of said second clamp section so as to force said second clamp section to radially compress said bushing into locking engagement with said outer end of said steerer tube.

15. A bicycle fork and handlebar stem assembly comprising:

a fork steerer tube, an end portion of which has a cylindrical outer surface;

a handlebar stem having first and second compression connectors which fit around said fork tube end portion, said first compression connector having an inner surface that is sized to make a close fit with said fork tube end portion, said second compression connector having an inner surface that is provided with a first screw thread; and

a castellated bushing surrounding said fork tube end portion and extending into said second compression connector, said bushing having an outer surface that is provided with a screw thread for making a screw connection with said first thread;

said first and second compression connectors each comprising means for tightening said connector around said fork tube and said bushing respectively, whereby said stem is locked to said fork tube and said bushing is compressed into tight engagement with said fork tube.

16. A handlebar stem assembly for attachment to a bicycle fork steerer tube comprising:

a castellated externally-threaded bushing adapted to fit on a bicycle fork steerer tube; and

a handlebar stem comprising a body section and first and second split tubular clamp sections formed integral with said stem section, said first clamp section being adapted to surround and be compressed into tight locking engagement with a bicycle fork steerer tube, and said second clamp section surrounding and making a screw thread connection with said bushing and being adapted to be compressed so as to squeeze said bushing into tight locking engagement with said bicycle fork steerer tube.

17. An assembly according to claim **16** wherein said bushing comprises a castellated tubular body section and an exterior flange section at one end of said body section.

18. An assembly according to claim **16** wherein one end of said bushing projects out of said second clamp section.

19. A bicycle handlebar stem assembly comprising:

a handlebar stem having first and second compression connectors each adapted to extend around a fork steerer tube, with said second compression connector having an inner surface that is provided with a first screw thread; and

a castellated bushing sized to fit within said second compression connector, said bushing having an outer surface that is provided with a screw thread for making a screw connection with said first screw thread and an inner surface shaped to surround a fork steerer tube;

said first compression connector having means for tightening said first compression connector around a fork steerer tube that is surrounded by said first compression connector, and said second compression connector having means for tightening said second compression connector around said bushing.

* * * * *

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 6,186,027 B1
DATED : February 13, 2001
INVENTOR(S) : Peter M. Nielsen

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

Claim 7, column 6,
Line 4, change "stem" to -- body --;

Claim 12, column 6,
Line 43, change "stem" (second occurrence) to -- body --;
Line 46, change "stem" to -- body --;

Claim 13, column 6,
Line 59, change "stem" to -- body --;

Claim 16, column 8,
Line 7, change "stem" to -- body --.

Signed and Sealed this

Second Day of October, 2001

Attest:

Nicholas P. Godici

NICHOLAS P. GODICI

Attesting Officer *Acting Director of the United States Patent and Trademark Office*

US006381827B1

(12) **United States Patent**
Steinbock

(10) **Patent No.: US 6,381,827 B1**
(45) **Date of Patent: May 7, 2002**

(54) **METHOD FOR MAINTAINING A CLAMPING FORCE BETWEEN BOLTED PARTS IN HIGH TEMPERATURE**

(75) Inventor: **Rolf H. Steinbock**, Carnegie, PA (US)

(73) Assignee: **Steinbock Machinery Co.**, Carnegie, PA (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/663,283**

(22) Filed: **Sep. 15, 2000**

Related U.S. Application Data

(62) Division of application No. 09/067,587, filed on Apr. 28, 1998, now Pat. No. 6,199,453.

(51) **Int. Cl.**[7] **B21D 39/00**; B25B 17/00; F16B 31/00; F16B 27/00

(52) **U.S. Cl.** **29/452**; 81/57.38; 411/14.5; 411/916

(58) **Field of Search** .... 29/452, 525.02, 29/525.11, 256; 81/57.38; 411/14.5, 916, 411, 424; 254/29 A

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Assistant Examiner—John C. Hong
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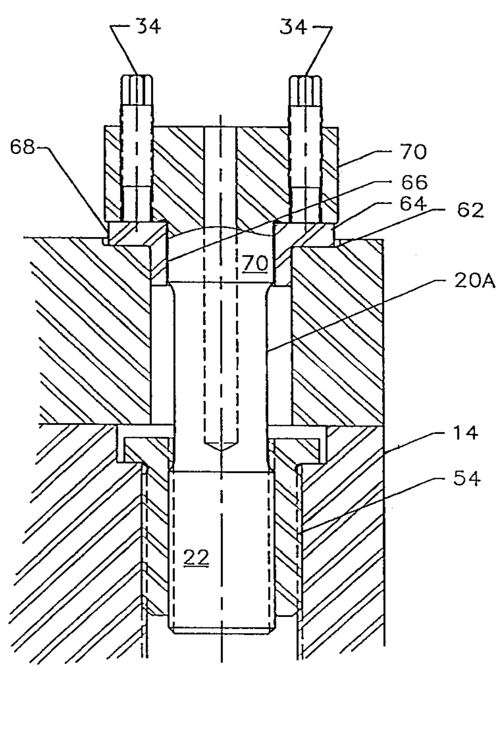
(57) **ABSTRACT**

A high temperature bolting system for clamping steam turbine housings and similar high temperature parts together while operating at a temperature of 800° F. to 1200° F. An elongated fastener shank and housing material have similar coefficient of thermal expansion. The shank has a tensile strength several times greater than the housing to maintain a clamping force during thermal excursions. The diameter of the shank is reduced due to the large creep strength difference between the elongated fastener shank and the clamped parts. The reduced shank diameter allows an increased in the clamping load by providing an increased surface area in the clamped part. A stress generating flange is provided with jackbolts to establish the clamping load at room temperature. A threaded sleeve is interposed between the threads at the end of the shank and the threads of the housing so that the housing threads have a sufficiently large area for withstanding the clamping load. A truncated conical internal wall in the shank in combination with a differential thread pitch in an embodiment provides higher resistance to the clamping force. A flanged support ring is mounted at a bolt hole in the flange to support a load bearing area when retrofitting existing housings.

17 Claims, 5 Drawing Sheets



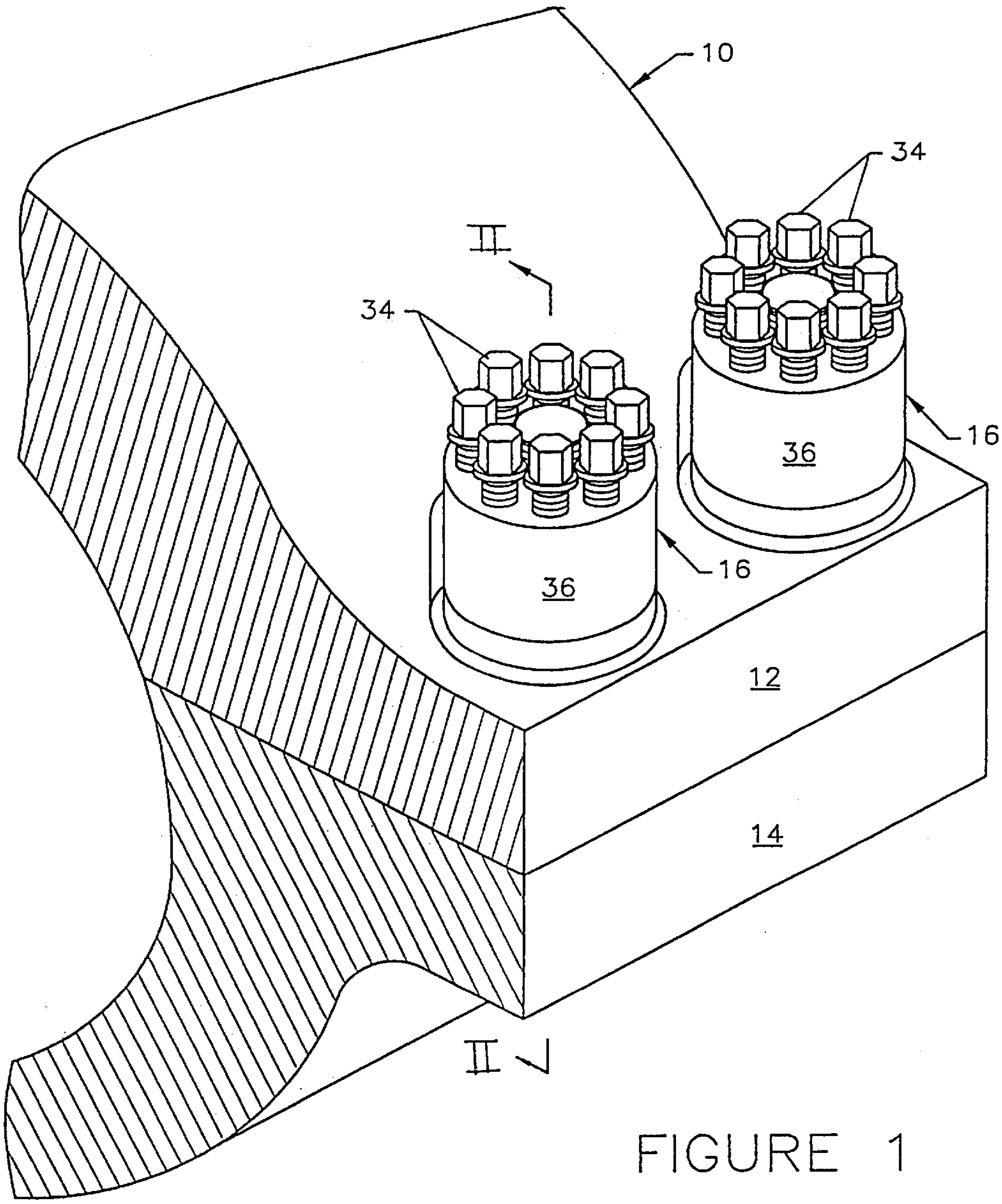


FIGURE 1

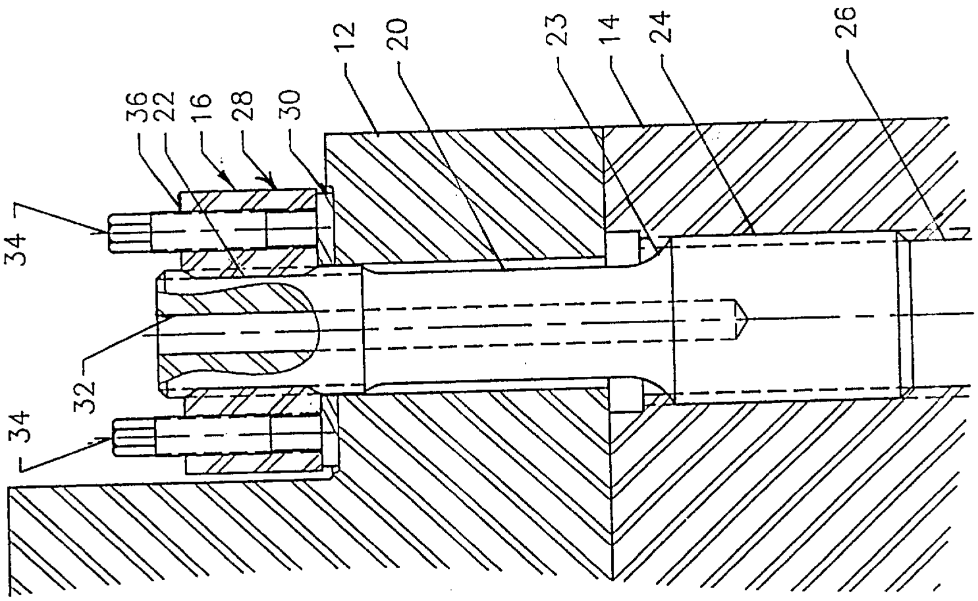


FIGURE 2

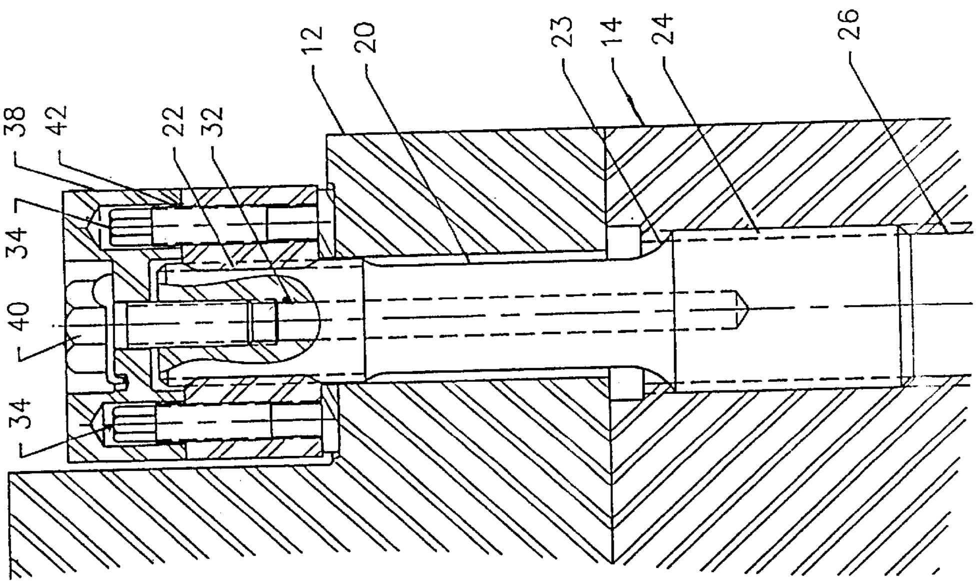


FIGURE 3

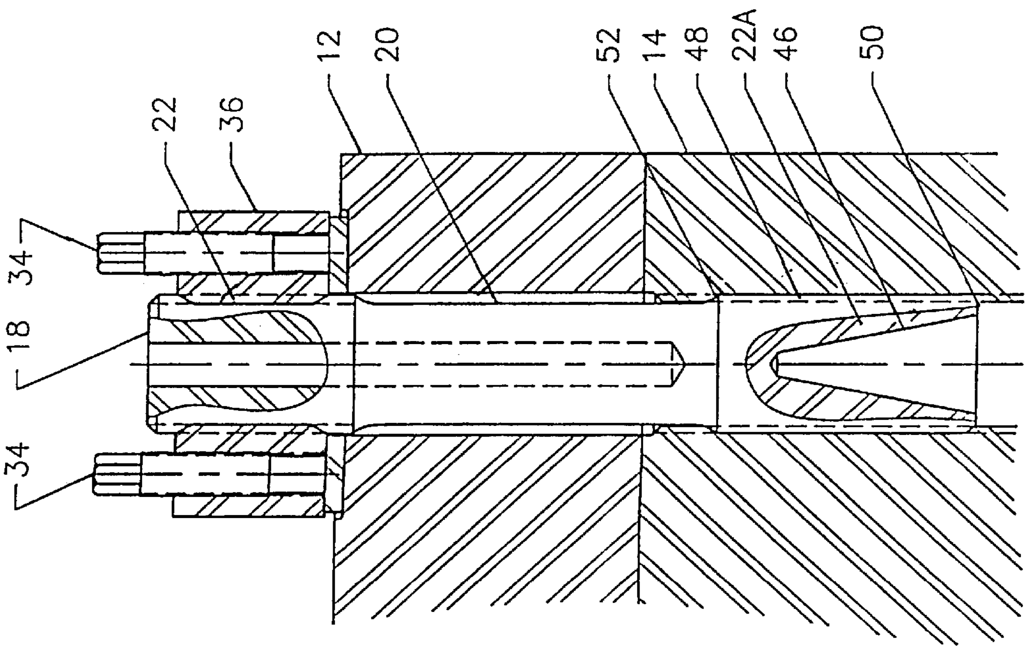


FIGURE 4

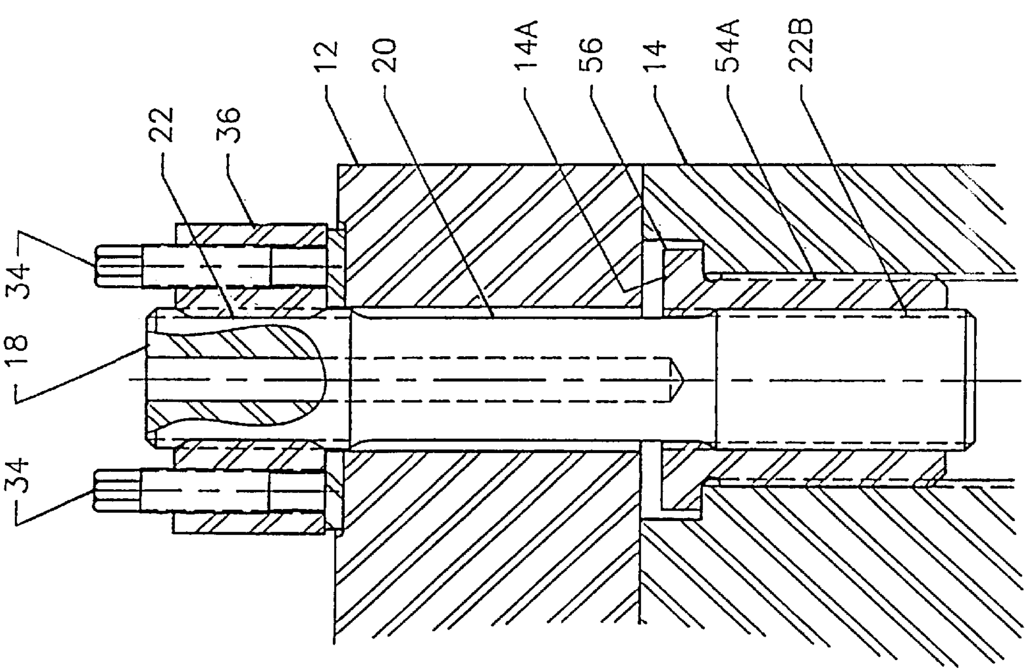


FIGURE 5

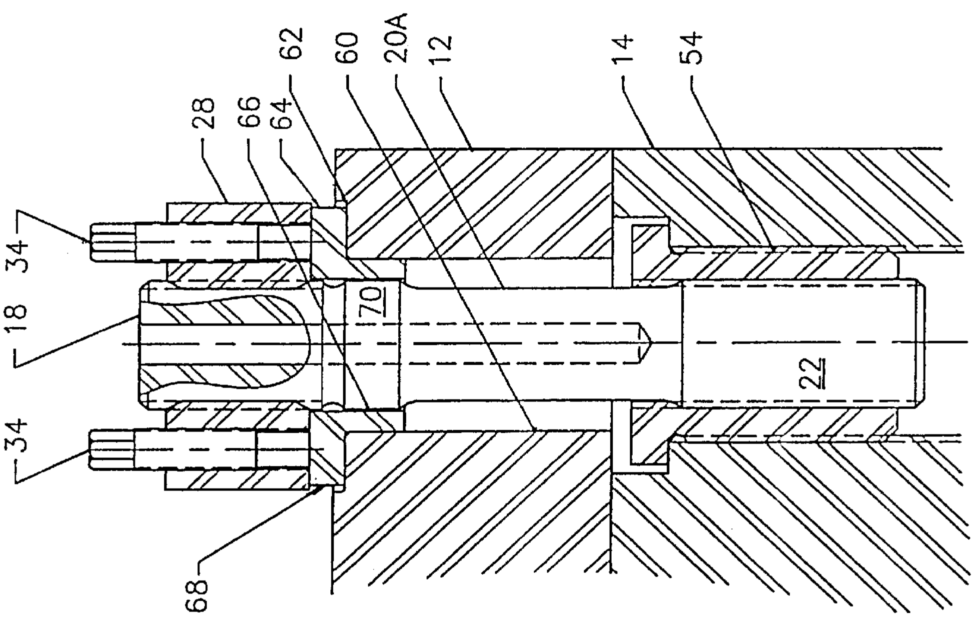


FIGURE 6

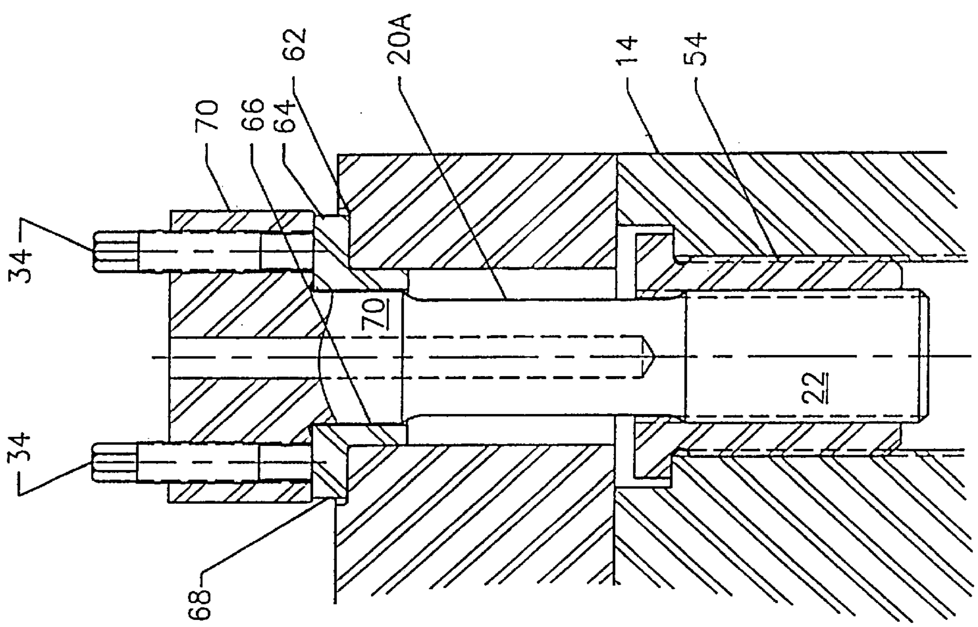


FIGURE 7

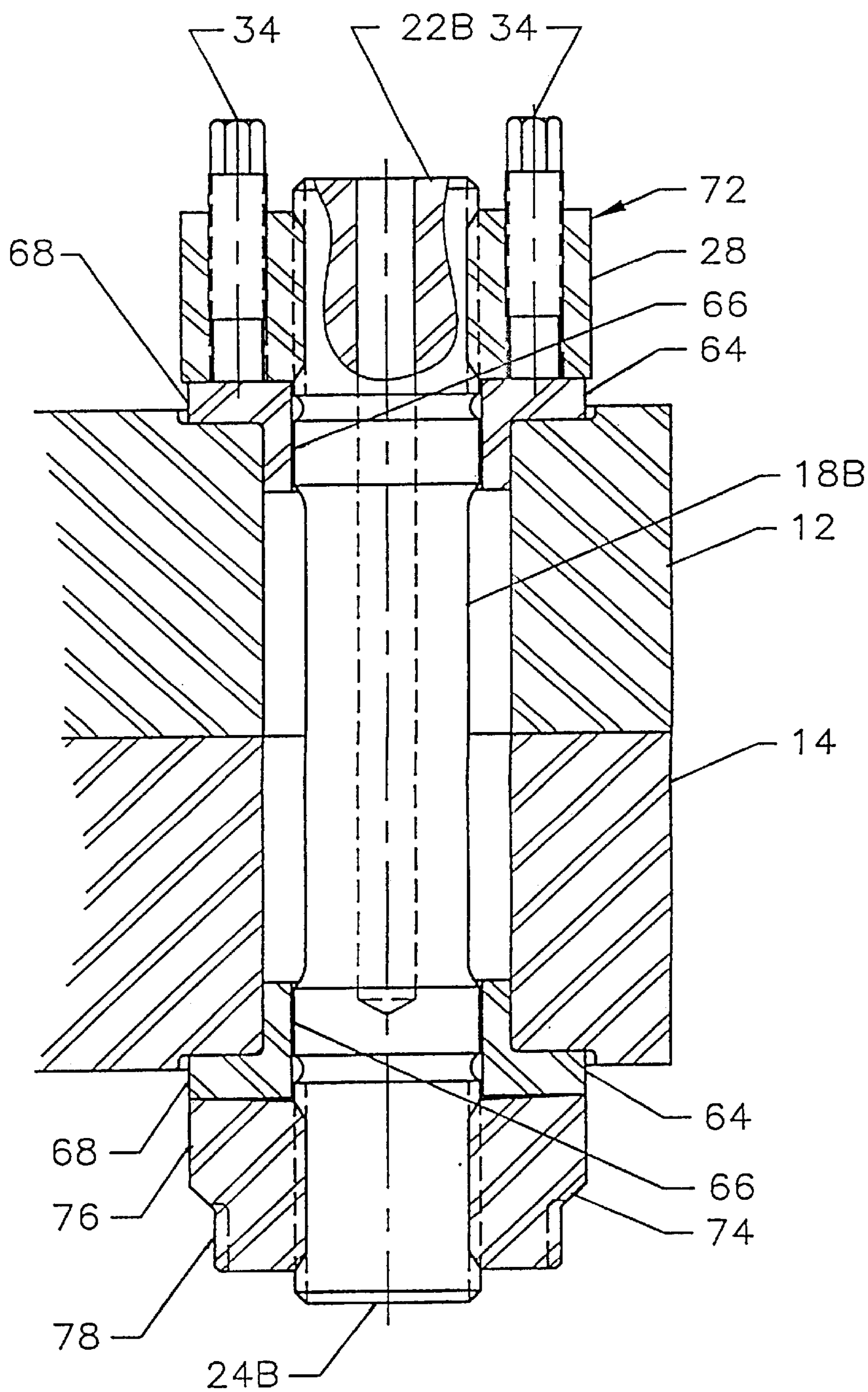


FIGURE 8

METHOD FOR MAINTAINING A CLAMPING FORCE BETWEEN BOLTED PARTS IN HIGH TEMPERATURE

RELATED APPLICATIONS

This application is a division of Ser. No. 09/067,587 filed Apr. 28, 1998, now U.S. Pat. No. 6,199,453.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a bolting system for maintaining a clamping force between bolted parts while operating at a temperature of 800° F. to 1200° F. and more particularly to a fastener construction providing a coefficient of thermal expansion similar to the coefficient of thermal expansion of the bolted parts for minimizing differential dimension changes due to temperature variations and providing a shank portion of the fastener with an elastic limit sufficiently great to maintain a clamping force between the bolted parts throughout a temperature range from ambient to the operating temperature.

2. Description of the Prior Art

The present invention is addressed to solving complex problems involving bolting systems required to operate at high temperatures for an extended operating time. Conventional bolting systems used to form high temperature joints, used particularly in steam turbines and similar joints, have the acute disadvantage that they become considerably weaker when the operating temperature increases. It is necessary in such a bolting system that components are designed to clamp the joint firmly without damaging the most expensive part, the main turbine casting. In some cases the clamped part may be an expensive valve body or any other component. While the present invention is not so limited, a bolting system for a main steam turbine casing has been chosen for the purpose of disclosing the present invention.

At elevated temperatures of the order of 800° F. to 1200° F., bolting parts, typically studs tensioned in a threaded tapped hole, often fail because of an improper selection of stud material. The studs soften, stretch and become loose. Other problems occur because the stud is subjected to differential expansion causing the clamped part to become loose when the stud expands faster than the clamped part and similarly the stud becomes over stressed when stud expansion is slower than the clamped part. In the latter circumstance conventional studs stretch permanently under the clamping force because the force is too great.

By way of a detailed example of such bolting problems, is the bolting of the split casing of a steam turbine. The casing in the area around the steam inlet reaches a temperature typically of 1000° F. to 1050° F. The turbine casing is usually heavy with wall thickness ranging from two to eight inches thick formed by casting a chrome molybdenum vanadium steel. This cast steel is relatively strong at 1000° F. compared to other low cost casting materials. Relatively strong means it will soften at 1000° F. to less than 10,000 psi creep strength while other low cost castings soften to 2,000–5,000 psi creep strength at 1000° F. Most casting materials start at about 40,000 psi to 60,000 psi yield strength when cold. Steam turbines have to withstand not only high temperatures, but also high pressures. The high temperatures, high pressures and relatively soft materials demand that the castings are made with very thick walls. The larger the turbine the thicker the wall.

Since a turbine housing is split along its axis, the two halves have to be bolted together. On large turbines the bolts may be as large as 6 inches in diameter. On small turbines the bolts may be 2 inches in diameter. In many cases the bolts are made from a material similar to the material of the housing, which in turn allows both, bolt as well as housing, to expand at the same rate as the temperature rises. This has been the practice for many years when the chrome molybdenum vanadium steel casting was bolted together with chrome molybdenum vanadium steel bolts. By design the combined cross section of the chrome molybdenum vanadium steel bolts is smaller than the cross section of the cast housing. After a certain time the bolts relax to a point where they do not clamp the two castings sufficiently causing the turbine to leak steam. Turbine designers try to estimate the time it takes for the bolts to relax sufficiently to cause leaks. A five year period was usually considered the time span before leaks occurred and corresponds to scheduled turbine overhaul. However, the estimated time period was not always accurate. Presently, operators prefer to extend the time between overhauls and also raise the operating temperature by up to 50° F. to improve efficiency.

To solve the above problems turbine designers and operators attempted to improve performance by using substitute bolting materials. Some improved performance was obtained by the addition of trace elements such as boron to the basic chrome molybdenum vanadium steel, but the improvements obtained were not sufficient. Many substitute materials were strong enough to operate at the required high temperatures but the expansion coefficient of the bolting material did not match that of the steel casting material. Many heat resistant stainless steels expand at a rapid rate causing a reduction and in extreme circumstances an actual loss of the clamping force provided by the tightening process when reaching the operating temperature of the turbine. The following example demonstrates the problem encountered due to differential expansion at high temperature between bolting material and the turbine casting:

Example:

Coefficient of thermal expansion (turbine housing)	7.9×10^{-6} Inch/° F./Inch
Thickness of housing flange	10 inches
Operating temperature - Room temperature	1000 − 70 = 930° F.
Expansion of flange:	$7.9 \times 10^{-6} \times 10 \times 930 =$ 0.073 inches
Coefficient of expansion (high temp. bolt A286)	9.8×10^{-6} Inch/° F./Inch
Expansion of bolt	$9.8 \times 10^{-6} \times 10 \times 930 = 0.091$ inches
Difference in expansion	0.091 − 0.073 = 0.018 inches

This example demonstrates that the expansion of the bolt exceeds the expansion of the flange by 0.018 inches. Tensioning the bolt to create a preload will not resolve the problem. Expansion of the bolt, because of the preload, at room temperature produces an elongation according to Hook's Law:

bolt length	10 inches
bolt stress	45,000 psi

-continued

modulus of elasticity	30×10^6 psi

The bolt was stretched 0.015 inches by the tension causing the preload but the bolt expanded 0.018 inches due to thermal expansion from 70° F. to 1000° F. Thus the bolt is now loose by 0.003 inches which is obviously not a workable solution.

Another bolting material was found to be unsuitable and that is a martensitic stainless steel of the 400 series. The addition of vanadium to the basic alloy creates an alloy with about twice the strength at high temperatures as compared with the chrome molybdenum vanadium bolting material. The improved 400 series stainless steel has AISI designation of 422 and is known in the industry as "12% chrome steel". This chrome steel used as a bolting material in Example 1, having a thermal expansion coefficient of 6.4×10^{-6}, results in the expansion of the 10 inches bolt by $6.4\times10^{-6}\times10\times930=$ 0.059 inches. The expansion of the flanges of the housing is 0.073 inches and therefore the bolt must stretch 0.014 inches in excess of the 0.015 inches resulting from the tightening procedure. The total stretch is now 0.015 inches+0.014 inches=0.029 inches. Using Hook's law the tension in the bolt would now be:

$$0.029\times30\times10^6/10=87{,}000 \text{ psi}$$

The stress increased by a factor of about 2. In reality Hook's law does not apply when materials are stressed above their elastic limits. The 12% chrome material is much too soft at 1000OF to sustain the 87,000 psi stress. Under these operating conditions the 12% bolting material can sustain only about 18,000 psi of stress over long periods (10,000 hours or more). When over-stressed, the bolt will stretch permanently and therefore loose the tension producing the preload. Another problem arises when the joint of the housing cools down. The casting will shrink faster than the bolt (caused by the different coefficients of expansion) and the bolt will become loose. It will have to be re-tightened before the turbine can be started up again. During the following operation of the turbine the high temperature will stretch the bolt again. Soon the bolt has stretched to such an extent that the threads no longer fit the threads of the nut and a replacement is necessary. Another problem with 12% chrome steel is that the material becomes brittle and cracks during the use. A failure of more than one bolt can lead to catastrophic consequences of the steam turbine.

A nickel super alloy such as Inconel 718 has the desired properties for bolting materials required to operate at temperatures of 1000–1050° F. but a completely new approach to the bolted joint design is needed to successfully operate in the high temperature and high pressure environment in the operating steam turbine. Nickel super alloys will not significantly soften at high temperatures and these materials are approximately 10 times stronger at 1000° F. than the chrome molybdenum vanadium steel and approximately 5 times stronger than the 12% chrome steel. The super alloy has a coefficient of thermal expansion closely matching the coefficient of thermal expansion of the turbine housing casting.

Because of the strength of the material, the design of the fastener must address a whole new set of problems. One particular problem is the method of tightening the fastener to provide the desired clamping force at both ambient and operating temperatures. Thermal tightening is not reliable and large preloads are difficult to obtain. Studs in turbine casings are spaced very closely and hydraulic tensioners cannot be used due to insufficient space and for the same reason hydraulic wrenches are not suitable. Another serious problem when using super alloys is that the extremely strong bolts can now deform the turbine casting. Assume that a turbine is heated to 1000° F. and the bolt was tightened to 45,000 psi. Due to the same expansion coefficient of the bolt and housing the clamping force does not change. The bolt is strong, it has a creep strength of 100,000 psi even at 1000° F., but at 1000° F. now the housing has a creep strength of less than 10,000 psi. Something has to give, in this case it is the housing. The 100,000 psi bolt can easily deform the 10,000 psi housing causing serious damage.

It is an object of the present invention to provide a bolting system that will maintain a predetermined clamping force while in use at elevated temperatures in the range of 800° F. to 1200° F. and more particularly at a typical temperature of 1000° F. over a long time period without damage to the bolted parts.

It is a further object of the present invention to provide a fastener having a stress releaving threaded joint with the material of the bolted part for transferring stress from the fastener to the relatively low strength material of the bolted part without damage to the threaded connection in operating conditions of extreme thermal excursions.

It is another object of the present invention to provide a fastener system featuring a large load bearing support for a bolted part transverse to the path of stress in the fastener to avoid upsetting the material of the bolted part in operating conditions of extreme thermal excursions.

SUMMARY OF THE PRESENT INVENTION

According to the present invention there is provided a method for maintaining a clamping force between bolted parts while operating at a temperature of 800° F. to 1200° F., the method including the steps of selecting an elongated fastener having a stress generator and a connector at opposite ends of a shank portion, the shank portion having a coefficient of thermal expansion similar to the coefficient of thermal expansion of such bolted parts to minimizing differential dimensional changes due to temperature variations, the selected shank having an elastic limit sufficiently great for maintaining a clamping force between such bolted parts throughout a temperature range from ambient temperature to an operating temperature of between 800 degrees F to 1200 degrees F, the selected fastener shank having a creep strength several times greater than the creep strength of such bolted parts at the operating temperature; installing the selected fastener to mechanically join such bolted parts, and operating the stress generator substantially at an ambient temperature to stress the shank part sufficiently within elastic limits thereof to maintain a clamping force at the operating temperature of between 800 degrees F to 1200 degrees F.

BRIEF DESCRIPTION OF THE DRAWINGS

These features and advantages will be better understood when read in light of the accompanying drawings in which:

FIG. **1** is a fragmentary isometric view of a turbine casing incorporating the high temperature bolting system according to the present invention;

FIG. **2** is a sectional view taken along lines II—II of FIG. **1**;

FIG. **3** is a sectional view similar to FIG. **2** and illustrating a further embodiment of the present invention;

FIG. **4** is a sectional view similar to FIG. **2** and illustrating a further embodiment of the present invention;

FIG. **5** is an illustration similar to FIG. **2** of a further embodiment of the present invention particularly useful for providing fasteners to existing turbine casings;

FIG. **6** is a view similar to FIG. **5** and illustrating a further embodiment of a fastener for existing turbine casings;

FIG. **7** is a view similar to FIG. **5** and illustrating a further embodiment of a fastener for existing turbine casings; and

FIG. **8** is a view similar to FIG. **5** and illustrating a thru-bolt system according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

A portion of a steam turbine **10** is illustrated in FIG. **1** and includes mating flanges **12** and **14**, typically made of cast chrome molybdenum vanadium steel. Each flange is of a substantial thickness, typically ranging from 2 inches to 12 inches or more depending on the site about the periphery of the steam turbine. As set forth hereinbefore the coefficient of thermal expansion produces an expansion of the 10 inch flange of 0.073 inches. Side by side fastening systems **16** are provided to bolt the flanges together under a clamping force that is sufficiently great so that the required clamping force remains operative throughout a temperature range from ambient temperature to an operating temperature of between 800° F. and 1200° F.

According to the various embodiments of the present invention, a stud or threaded bolt made from Inconel 718 or another super alloy with an expansion coefficient similar to the housing material, is selected to clamp the chrome molybdenum vanadium mating flanges of the steam turbine **10**. In the embodiment illustrated in FIG. **2**, the fastening system **16** includes a stud **18** having an elongated fastener shank **20** with threaded portions **22** and **24** at opposite ends of the shank. The shank is stepped, identified by reference numeral **23**, to provide a reduced diameter along the length thereof as compared to the pitch diameter of the threads in threaded portion **24**. The stepped configuration allows a large shear area where the stud is threadedly engaged with internal threads **26** in flange **14**. Because the material of the flange is relatively soft the large shear area prevents stripping of the threads. Mating engagement between threads **24** and **26** form a connector at one end of the fastener shank and at the opposite end of the fastener shank a stress generating flange **28** is in mating engagement with threads **22** of the fastener shank.

The reduced diameter of the fastener shank **20** provides a larger footprint for seating engagement with a hardened washer **30**, as compared with the footprint of the original larger diameter stud washer. The enlarged footprint is necessary to distribute the clamping force to a greater support area of the flange **12**. The large foot print is achieved by enabling a smaller inside diameter and a larger outside diameter for the washer **30**. The outside diameter of washers for high temperature bolting systems should be as large as the space permits. A gain of 1/16 inch or more can substantially reduce stresses under the washer. Another benefit provided by the reduced diameter of the fastener shank is, that at the same tightening force a higher stress in the stud **18** is produced thereby providing more elongation and more elasticity. The increased elasticity provides more resiliency to elastically withstand the thermally generated stresses. The chrome molybdenum vanadium steel has a coefficient of thermal expansion of 8.0^{-6} inch/° F./inch which is sufficiently similar to the coefficient of thermal expansion of 8.0^{-6} inch/° F./inch for the Inconel 718 forming the fastener shank **20** to provide the features and advantages of the present invention. As will now be apparent to one skilled in the art, the cast steel material of the flanges **12** and **14** have a creep strength at an operating temperature of 1000° F. of 10,000 psi and are protected against elastic deformation given that the bolting shank material has a comparable creep strength, Inconel 718, of at least 100,000 psi at 1000° F. As can be seen, the creep strength ratio is 10:1 although the ratio can range from 4:1 to greater than 10:1 particularly with the development of special purpose alloys. Thus, the alloy Inconel 718 can withstand a mechanical stress of at least 108,000 psi at 1000° F. and can easily withstand the higher stresses without risking plastic deformation.

Because of the extreme stress represented by the clamping force, it is difficult to loosen the stress generated flange after heating when all lubricants have evaporated. To reduce loosening torque it is preferred to relieve at least some of the stress by a thermally induced elongation produced by introducing a low cost cartridge heater into the internal body of the shank through a hole **32** drilled into the center of the stud. The heating element will be necessary only in severe instances to facilitate loosening a tight stress generated flange. Timely lubrication of the threads at the stress generating flange and quick heating of the stud **18** through the center hole **32** will lower the required break loose torque.

The stress generating flange **28** is preferably in the form of a multi-jackbolt tensioner which forms an important part of the preferred embodiment of the high temperature bolting system of the present invention. The stress generating flange is of the general type disclosed in U.S. Pat. No. RE. 33,490. As shown in FIGS. **1** and **2**, a plurality of jack bolts are threadedly engaged in tapped holes at spaced apart locations about the outer periphery of a tensioner body **36**. Preferably, as shown, the tensioner body **36** takes the form of a circular ring. This form of a tensioner body provides increased strength without increasing the outside diameter from that of the outside diameter of a standard hexagon shaped nut. An important concept of the present invention is to increase the load bearing area by which the washer **30** transmits a compressive force to the flange **12**. The present invention takes advantage of the property of the casting material used in turbines and similar high temperature applications which is 40% stronger in compression than tension. The elongated fastener shank comprised of a nickel based super alloy is constructed with a reduced diameter as compared with a conventional shank diameter to allow more stretch to the shank under the varying load conditions imposed by the thermal expansion. With superalloy fasteners the shank always operates within the elastic limits of the material and maintains the desired clamping pressure. By reducing the diameter of the shank, the threaded portion **22** is correspondingly reduced so that the inside diameter of washer **30** is reduced thus increasing the area of the load bearing face of the washer in contact with flange **12**. This relationship allows the same pounds per square foot loading by the washer **30** but a much larger load can be transferred because of the increased area of the load bearing face of the washer.

The following example demonstrates this benefit of the present invention. A 2 inches nominal diameter bolt shank will be used with a washer having a 2 inches inside diameter and a 3 inches outside diameter thereby defining a load bearing face area of 3.9 sq. inches. The fastener of the present invention can be provided with a shank having a diameter of 1.5 inches whereby the inside diameter of the washer can be nominally 1.5 inches and the outside diameter 3 inches. This provides a load bearing area of 5.3 sq. inches. To transfer the loads from the jack bolt without upsetting the material of the washer, the washer is made from a super

alloy such as Inconel 718, A 286 or other high temperature nickel based super alloys. Because of the relatively small washer thickness, the thermal expansion coefficient of the washer material does not materially affect the bolting system. The tensioner body is significantly strengthened to withstand the increased clamping pressure because the threaded end portion **22** of the tensioner has a smaller diameter allowing a larger, stronger tensioner body **36** with larger diameter tapped holes for the jackbolts **34**. One suitable material for the nut body is 316 stainless steel (ASTM A194-8M) to provide sufficient strength at the high operating temperatures. Jackbolts are made from super alloys, Inconel 718,is a preferred jackbolt material, because it can be used with the nut body made from 316 stainless steel to prevent seizure and galling of the jackbolts in the tensioner body **36** under load. The relatively high thermal expansion coefficient of the tensioner body **36** as compared to the expansion coefficient of the fastener shank **20** is of little consequence since the nut body is relatively short in length and will expand only to a small relative extent. The treaded portion **24** is formed with an enlarged diameter as compared to the diameter of the fastener shank **20** and threaded portion **22** to provide a necessary increase to load transfer area formed by the threaded connection with the internal threads in flange **14**. In the example discussed previously involving the use of a nominal 1½ inches diameter Inconel fastener, the shank material will have a strength factor of between 8 to 10 times greater than the strength of the material of flange **14**.

To demonstrate the advantage of utilizing the 40% greater strength of most metals in compression as compared with tension, the load bearing area of a standard high temperature washer (3" O.D. 2" I.D.) for a 2" bolt is 3.39 sq. in. The area of a standard high temperature 2" bolt (minor diameter 1.84") is 2.65 sq. in. The ratio of the area of the washer to the area of the bolt is 1.48. This ratio remains constant for all standard high temperature washer and bolt combinations. According to the present invention, the load bearing area of a washer (3" O.D. 1.5 I.D.) is 5.3 sq. in. The area of the super alloy bolt (minor diameter 1.34") is 1.4 sq. in. In the bolting system of the present invention, the ratio of the area of the washer to the area of the bolt is 3.79. This ratio will not change significantly for all new high temperature washer and bolt combinations according to the present invention. The use of nickel based super alloy for the bolting material enables the tremendous increase to the ratio which as can be seen provides the great benefit of an increase to the area for the transfer of load in compression and thereby maintain the loading per square inch by the washer within elastic limits of the metal in compression. In the standard high temperature bolting system, the minor area of the bolt in tension was the limiting factor as to the strength of the bolting system. In the new high temperature bolting system, the area under the washer which is larger than the minor area of the bolt in tension becomes the limiting factor. The new limiting factor is increased not only because of the significant increase to the washer area but also because the compressive load transfer has the advantage of a 40% greater capacity than load transfer in tension. In the example of a standard high temperature 2 inch bolt, a minor bolt area of 2.65 sq. in. with a creep strength of 7000 psi results in a load bearing capacity of: 2.65 sq. in.×7000 psi=18,555 lbs. The new bolting system the area under the washer is 5.3 sq. in. with a creep strength of 7000 psi and the benefit of a compression factor of 1.4 results in a load bearing capacity: 5.3 sq. in.×7000 psi×1.4=51,940 lbs. The significant increase is really achieved by the improved bolting system of the present invention.

Many steam turbines have two casings, an inner casing and an outer casing. The inner casing and the bolting for the inner casing are exposed to live steam. As shown in FIG. **3**, to protect the exposed torque receiving ends of the jackbolts **34**, a heavy metal cap **38** is attached to the stud **18** when the stud is a discrete member or when the stud is integral with the tensioner body **36**. The cap **38** has a central aperture to allow the threaded portion of a bolt **40** to pass through the cap and into a threaded hole in the end portion of the fastener shank. Discrete cavities **42** are formed in the cap to protectively house the exposed end portions of each of the jackbolts **34**. An annular cavity in the cap **38** can be used in place of the discrete cavities to protect the jackbolts. In this way the multi-jackbolt tensioners are protected from contact with live steam. Attaching the cap to a stud is relatively easy on super alloy studs when the stud is drilled to form the heater hole **32**. It is only necessary to tap the top of the heater hole to receive the threads of a bolt **40**. A lock washer **44** is preferably provided to secure the bolt against unwanted release from tapped hole. In the embodiment of FIG. **3**, the remaining parts and relationship between the parts are the same as described in regard to FIG. **2**.

The high strength of the nut body 36 is advantageously utilized by converting the large diameter threaded portion **24** to an extended length with a nominal diameter which can be substantially the same as the diameter of the shank **20**. As shown in FIG. **4**, the threaded portion **22A** is constructed with an internal cavity defined by a truncated conical wall **46** which serves as an elastic element to evenly distribute thread forces along the length of the threaded portion **22A** up to the extreme operating temperatures. A tapered thread over the length of the threaded portion **22A** creates a differential thread pitch to the threads **48**. The effect produced by the differential thread pitch moves stress concentrations in the threads **48** to the terminal end portion **50** toward the tensioner body from a site **52** which is close to the origin of the fastener shank **20**. The creation of the differential thread pitch distributes the load imposed by the stress generating flange **28** more evenly along the extended length of the threads **48** and thereby avoid over stressing of the threads within a small area.

Another embodiment of the present invention is illustrated in FIG. **5** and provides a threaded bushing **54** which due to its large outside diameter is used to distribute the forces generated by the stress generating flange safely to the flange **14** without the danger of damage to the threaded connections. The bushing **54** is preferably formed from the same material as the nut body, namely, stainless steel and thereby act in unison with the nut body to maintain the clamping force generated by the fastener on the mating flanges **12** and **14**. As can be seen in FIG. **5**, the diameter of the threaded portion 22B extending from the fastener shank **20** is approximately the same as in the example given above the nominal diameter of the threads may be 1½ inches. The external threads **54A** on the bushing **54** terminate at a relief area where a shoulder **56** is provided to abut against a counterbore seat **14A** in flange **14**.

The present invention also allows retrofitting the bolting systems for the casing of existing steam turbines. retrofitting has presented challenging engineering problems because of the extreme temperature excursions and the existing dimensional limitations imposed by the existing design of the flanges of the casting. The flange of the top casing usually has a relatively large diameter hole to allow free passage of a stud threadedly engaged in the tap hole of the flange of the bottom casing. According to the present invention, the large diameter stud is replaced by a much

smaller diameter super alloy stud and as shown in FIGS. **6** and **7**, a relatively large annular gap is formed between external diameter of the fastener shank **20**A and the annular surface **60** defining the opening in flange **12**. Because of this relationship between the diameter of the fastener shank and the diameter of the hole in which the shank resides, a relatively small area is defined by a machined surface **62** in the exposed face of the flange to support the load transferred by the stress generating flange **28**. If a flat washer is used to transfer the En stresses generated by the jackbolts, the danger of deformation of the hot and rather soft housing metal is great and will likely occur. To enable transfer of the necessary forces generated by the stress by the fastener, a flanged support ring **64** is interposed between the stress generating flange **28** and the surface **62**. The ring **64** has the form of a circular ring portion **66** with an external annular flange **68** protruding from the circular ring portion. As shown, the annular flange **68** extends between flange **12** and the nut body **28** and serves to transfer the load from the jackbolts to the flange **12**. The annular flange **68** is constructed to tightly fit along a length of the annular opening **60** adjacent the machine surface **62**. The flange supplies the needed back-up support for the metal of flange **12**.

The backup support provided by this arrangement of parts serves to prevent the annular wall **60** of the housing from collapsing under the stress. The metal under the annular flange **68** is confined in the same manner that a hydraulic fluid is confined in a piston and cylinder assembly. The confined metal of the flange thereby provides a greater load bearing capacity than could be sustained if the metal was not confined. The flanged spacer is preferably made from a super alloy such as Inconel 718 to avoid possible destruction of the spacer under the forces imposed transverse to the elongated length of the fastener shank and the clamping force generated by the stress generating flange **28**. It will be observed that the bolting arrangement in FIGS. **6** and **7** provided the reduced diameter for the shank portion **20**A made of the superalloy metal as discussed hereinbefore to provide an increase to the area along the top surface of the flange **64** which receives the load from the jackbolts. In both FIGS. **6** and **7**,the embodiments of the present invention provide for the use of threaded bushing **54** to enable the safe distribution of the clamping force from the stress generating flange **28** to the flanges **14**. The embodiments of FIGS. **6** and **7** differ in that in FIG. **6** the stress generating flange **28** is discrete from the fastener shank **20**A. In FIG. **7**, the stress generating flange is identified by reference numeral **70** and is an integral part of the fastener shank **20**A.

A stud and nut as explained above, can be replaced by a through-bolt system reference numeral **72**. As shown in FIG. **8**, the exposed sides of the housing flanges **12** and **14** receive the same compressive loading by the stress generating flanges **28** threadedly engaged with threaded end portion **22**B of stud **18**B and a threaded nut **74** threadedly engaged with threaded end portion **24**B. To create the largest possible footprint for the opposing nut **72** a flanged support ring **64** is provided between each of the flanges **12** and **14** and flange **28** and nut **74**,respectively to prevent crushing of the relatively soft metal used to form the housing flanges. It is suitable to use a flexible nut design according to the disclosure by U.S. Pat. No. 4,846,614, which disclosure is incorporated by this reference thereto, to transfer the forces generated to a flange generated by the tensioner body **36** to the other flange. Essentially, the threaded nut **74** is constructed with an enlarged bottom portion **76** where contact with the flanged support ring **64** and a nut end portion **78** with a reduced diameter as compared to the diameter of portion **76**. The nut end portion **78** because of its reduced diameter can flex toward the internal threads in the nut. The load bearing face of the bottom portion **76** is provided with a relief area adjacent the internal threads whereby portions **78** flexes outwardly under load. The inward flexing of portion **76** and outward flexing of portion **78** mechanically creates a differential thread pitch through the nut causing an increase to load bearing contact in the threaded connection with the nut. The flanged support rings **64** function in the same manner as described hereinbefore. The thru-bolt system **72** is also useful for clamping flanges of a steam turbine having bolt hole openings of reduced diameter to closely match with only a small clearance to allow installing of stud bolts. In this circumstance, flat washers **30** will replace the flange of support ring **64** to allow distribution of the compressive force over a sufficiently great area and allow high temperature operation within the elastic limit of the metal of flanges **12** and **14**.

While the present invention has been described in connection with the preferred embodiments of the various figures, it is to be understood that other similar embodiments may be used or modifications and additions may be made to the described embodiment for performing the same function of the present invention without deviating therefrom. Therefore, the present invention should not be limited to any single embodiment, but rather construed in breadth and scope in accordance with the recitation of the appended claims.

I claim:

1. A method for maintaining a clamping force between bolted parts while operating at a temperature of 800° F. to 1200° F., said method including the steps of:

selecting an elongated fastener providing a sufficiently large shear area along a shank portion for preventing stripping of threads during high temperature service, the fastener having a stress generator and a connector at opposite ends of said shank portion, the shank portion having a coefficient of thermal expansion similar to a coefficient of thermal expansion of such bolted parts to minimizing differential dimensional changes due to temperature variations, the shank portion having an elastic limit sufficiently great for maintaining a clamping force between such bolted parts throughout a temperature range from ambient temperature to an operating temperature of between 800 degrees F to 1200 degrees F, the shank portion having a creep strength several times greater than the creep strength of such bolted parts at said operating temperature;

installing the selected fastener to mechanically join such bolted parts; and

operating the stress generator substantially at an ambient temperature to stress the shank portion efficiently within elastic limits thereof to maintain said clamping force at the operating temperature of between 800 degrees F to 1200 degrees F.

2. The method according to claim **1** wherein said step of operating the stress generator includes torquing jack bolts in threaded engagement with said stress generator at spaced apart locations about an outer peripheral portion of the stress generator.

3. The method according to claim **1** wherein the selected stress generator is a circular ring with a threaded central opening engageable with threads on an end portion of said shank portion and jack bolts in threaded engagement with said circular ring.

4. The method according to claim **1** wherein said step of selecting includes providing a threaded heat resistant bush-

ing with a coefficient of expansion greater than the coefficient of thermal expansion of said shank portion, said step of installing including using said threaded heat resistant bushing to interconnect said connector with one of said bolted parts.

5. The method according to claim **1** including the further step of installing an annular support between an aperture in one of the bolted parts and said shank portion for supporting an annular wall of the aperture against deformation.

6. The method according to claim **1** including the further step of installing an annular support between an aperture in each of the bolted parts and said shank portion for supporting an annular wall of the aperture against deformation.

7. A method for maintaining a clamping force between bolted parts while operating at a temperature of 800° F. to 1200° F., said method including the steps of:

selecting an elongated fastener having a stress generator and a connector at opposite ends of a shank portion, the shank portion having a coefficient of thermal expansion similar to a coefficient of thermal expansion of such bolted parts to minimizing differential dimensional changes due to temperature variations, the shank portion, having an elastic limit sufficiently great for maintaining a clamping force between such bolted pats throughout a temperature range from ambient temperature to an operating temperature of between 800 degrees F to 1200 degrees F, the shank portion having a creep strength several times greater than the creep strength of such bolted parts at said operating temperature, said stress generator having a circular ring with a threaded central opening engageable with threads on an end portion of said shank portion and jack bolts threadedly engaged with said circular ring, said fastener having a slightly tapered thread with the maximum diameter thereof at a terminal end portion of the fastener for creating a differential thread pitch between said elongated fastener and said one of such bolted parts whereby stress concentrations shift along said connector from a site proximate the shank portion of the fastener to the site of said maximum diameter of the tapered thread of the fastener, said fastener further providing a hollow cavity having a form of a truncated conical wall with the maximum diameter thereof at the terminal end portion of the fastener for providing elasticity to more evenly contribute contact stressed between a male and female threads;

installing the selected fastener to mechanically join such bolted parts; and

operating the stress generator substantially at an ambient temperature to stress the shank part sufficiently within elastic limits thereof to maintain said clamping force at the operating temperature of between 800 degrees F to 1200 degrees F.

8. The method according to claim **7** wherein said step of operating the stress generator includes torquing jack bolts in threaded engagement with said stress generator at spaced apart locations about an outer peripheral portion of the stress generator.

9. The method according to claim **7** wherein said step of selecting includes providing a threaded heat resistant bushing with a coefficient of expansion greater than the coefficient of thermal expansion of said shank portion, said step of installing including using said threaded heat resistant bushing to interconnect said connector with one of said bolted parts.

10. The method according to claim **7** including the further step of installing an annular support between an aperture in one of the bolted parts and said shank portion for supporting an annular wall of the aperture against deformation.

11. The method according to claim **8** including the further step of installing an annular support between an aperture in each of the bolted parts and said shank portion for supporting an annular wall of the aperture against deformation.

12. A method for maintaining a clamping force between bolted parts while operating at a temperature of 800° F. to 1200° F., said method including the steps of:

selecting an elongated fastener having a stress generator and a connector at opposite ends of a shank portion, the shank portion having a coefficient of thermal expansion similar to a coefficient of thermal expansion of such bolted parts to minimizing differential dimensional changes due to temperature variations, the selected shank having an elastic limit sufficiently great for maintaining a clamping force between such bolted parts throughout a temperature range from ambient temperature to an operating temperature of between 800 degrees F to 1200 degrees F, the selected fastener shank having a creep strength several times greater than the creep strength of such bolted parts at said operating temperature;

installing the selected fastener to mechanically join such bolted parts; operating the stress generator substantially at an ambient temperature to stress the shank part sufficiently within elastic limits thereof to maintain said clamping force at the operating temperature of between 800 degrees F to 1200 degrees F; and

heating an internal cavity extending trough said stress generator and along at least a part of said shank portion to relieve said clamping force for removal of the stress generating flange from such bolted parts.

13. The method according to claim **12** wherein said step of operating the stress generator includes torquing jack bolts in treaded engagement with said stress generator at spaced apart locations about an outer peripheral portion of the stress generator.

14. The method according to claim **12** wherein the selected stress generator is a circular ring with a threaded central opening engageable with treads on an end portion of said shank portion and jack bolts in threaded engagement with said circular ring.

15. The method according to claim **12** wherein said step of selecting includes providing a threaded heat resistant bushing with a coefficient of expansion greater than the coefficient of thermal expansion of said shank portion, said step of installing including using said threaded heat resistant bushing to interconnect said connector with one of said bolted parts.

16. The method according to claim **12** including the further step of installing an annular support between an aperture in one of the bolted parts and said shank portion for supporting an annular wall of the aperture against deformation.

17. The method according to claim **12** including the further step of installing an annular support between an aperture in each of the bolted parts and said shank portion for supporting an annular wall of the aperture against deformation.

* * * * *

US005584210A

United States Patent [19]

Gelbein

[11] **Patent Number: 5,584,210**

[45] **Date of Patent: Dec. 17, 1996**

[54] **HAND BRAKE LEVER ASSEMBLY**

[76] Inventor: **Mark Gelbein**, 24 Gun La., Levittown, N.Y. 11756

[21] Appl. No.: **502,853**

[22] Filed: **Jul. 14, 1995**

Related U.S. Application Data

[63] Continuation of Ser. No. 143,296, Oct. 26, 1993.

[51] **Int. Cl.**^{6} **F16C 1/10**; G05G 1/10

[52] **U.S. Cl.** **74/489**; 74/502.2; 403/110; 403/236

[58] **Field of Search** 74/488, 489, 502.2; 403/110, 234, 235, 236, 261, 373

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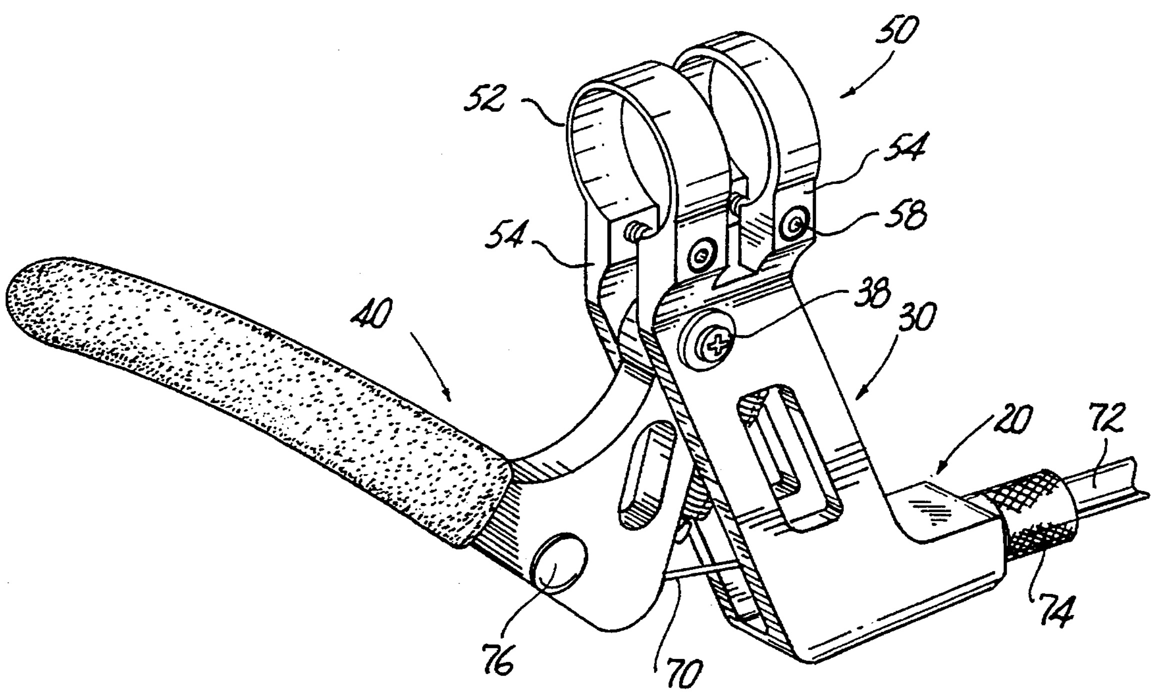
Primary Examiner—Charles A. Marmor
Assistant Examiner—Troy Grabow
Attorney, Agent, or Firm—Bauer & Schaffer

[57] **ABSTRACT**

A housing comprising a unitary block is bifurcated to form a pair of arms between which a crank lever is pivotally mounted. A circular clamp is simultaneously formed at the end of the arms to bridge the arms. The arms are adjustably drawn together to close the clamp. The crank lever is an L-shaped bell crank in which the pivot point and handle, when extended, intersect perpendicularly at a corner, forming the elbow about which the cable is pulled.

5 Claims, 2 Drawing Sheets



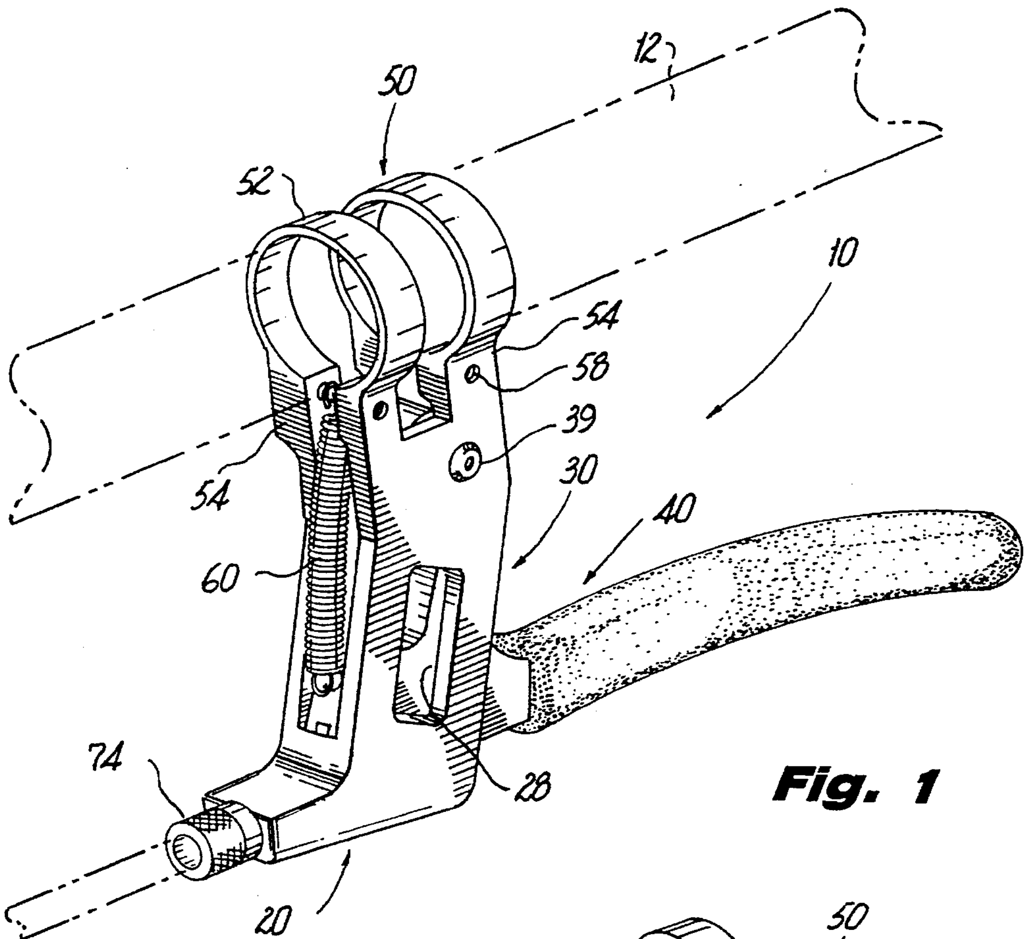

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Fig. 1

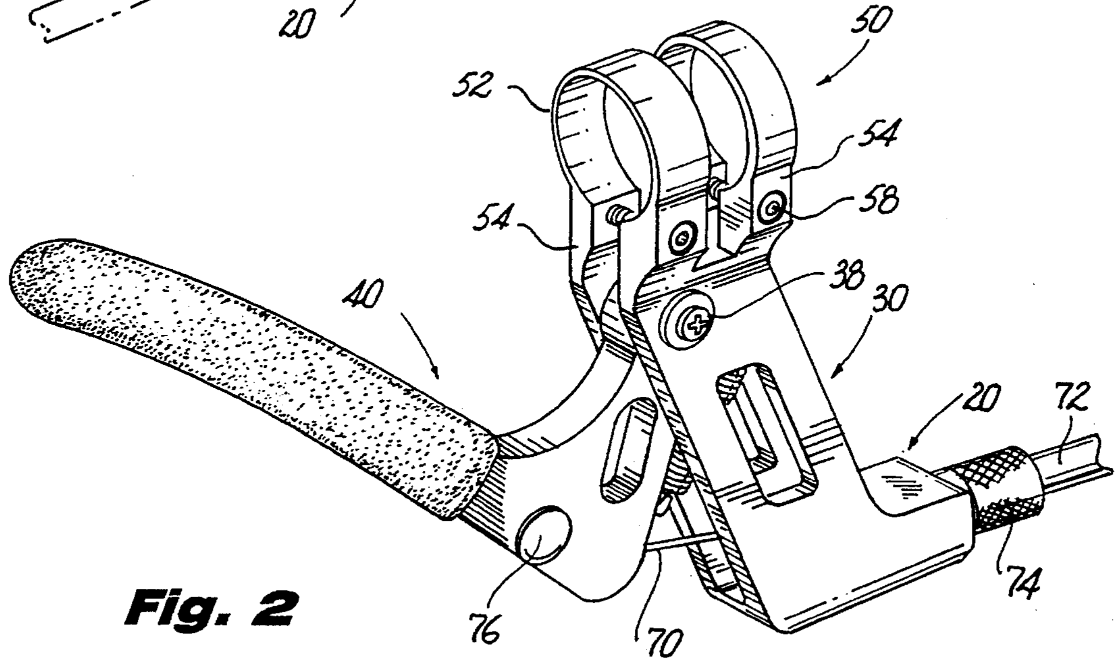
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Fig. 2

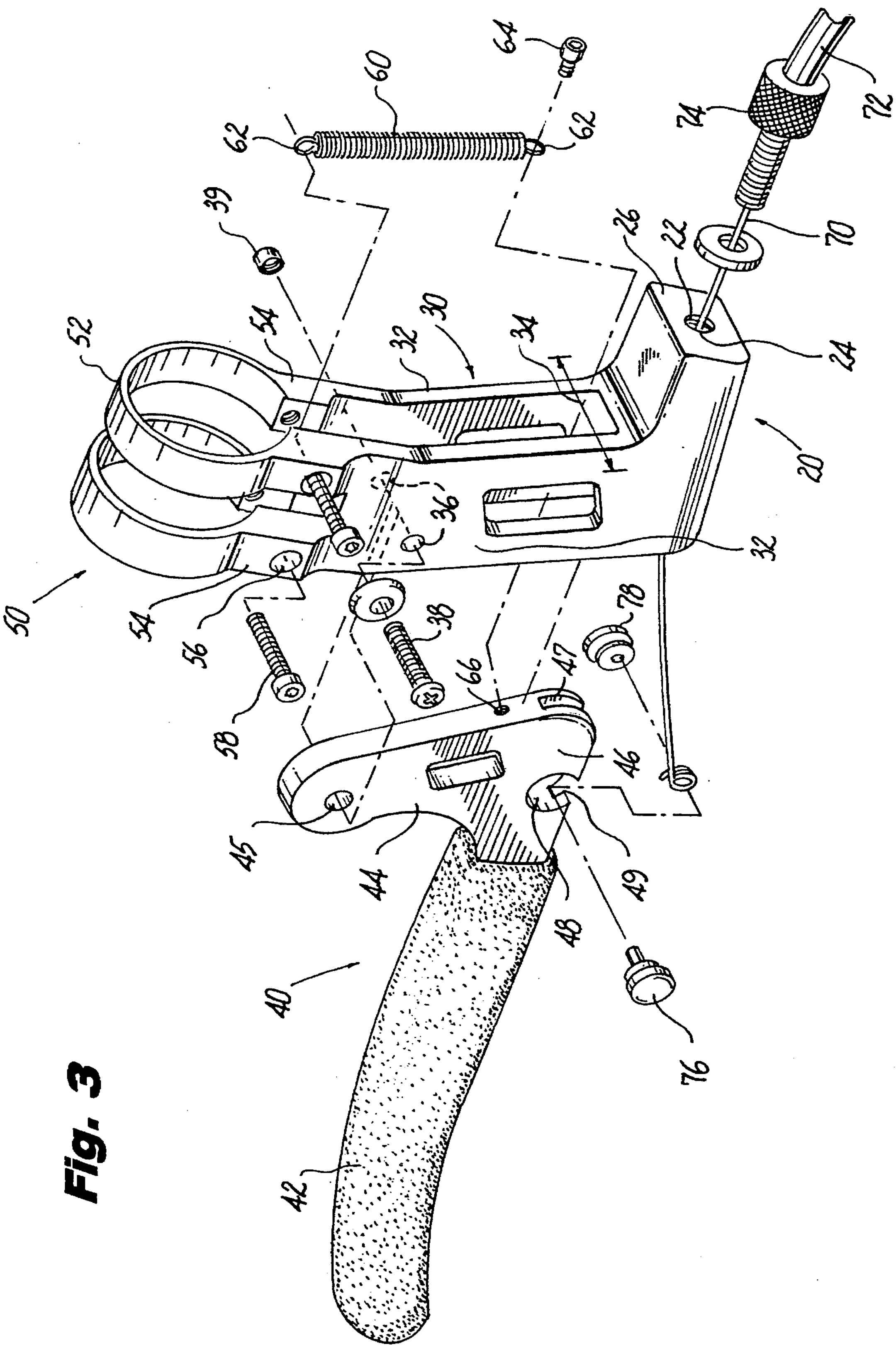
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Fig. 3

HAND BRAKE LEVER ASSEMBLY

This is a Continuation of Ser. No. 08/143,296, filed Oct. 26, 1993.

BACKGROUND OF THE INVENTION

The present invention relates to an improved hand brake lever assembly for bicycles.

Recent improvements in bicycles, and particularly front and rear derailleurs and gear shifting systems, have resulted in bicycles capable of enabling the biker to reach very high speeds over both smooth and rough terrain. Further, the use of esoteric metals have enabled the construction of lightweight, streamlined bikes having low air resistance, further allowing high speed racing. Nevertheless, one continuing problem exists in that the hand brake lever assembly, as commonly known and used, remains large, cumbersome, and weighty and, therefore, present an inordinate wind drag on the biker and bicycle. Another problem with the known lever assemblies is that they have relatively poor leverage action and, therefore, require relatively great strength to operate to arrest the forward movement of the bike.

An attempt has been made by several manufacturers to simplify the hand brake lever assembly. One such attempt is the Japanese, Shimano, BL-MT63 assembly. In this widely marketed assembly, the hand lever is pivotally mounted in a heavy monolithic aluminum housing. The housing is cast to form a pocket inwardly along one edge in which is mounted the lever and two heavy spring mechanism acting directly on the lever. As a result, the Japanese assembly is heavy, bulky, complex, and, therefore, far from being able to satisfy the modern needs of bikers.

In addition, a problem inherent in the Japanese assembly arises from the manner in which the handlebar clamp is fashioned. The clamp, in essence a heavy band, grows at one end out of the monolithic body and is bent so that its other end turns over the housing and is screwable into the housing. Because of the monolithic nature of the housing, the integral growth of the band from the housing and the need to screw the band at its only free end back onto the housing results in a poorly effective and weak clamp.

It is the object of the present invention to provide a bicycle hand brake assembly which overcomes the aforementioned problems and which provides an easily operable, lightweight hand brake having low drag when installed.

These objects, together with numerous other objects and advantages are set forth in the following disclosure of the present invention.

SUMMARY OF THE INVENTION

According to the present invention, a hand lever assembly is provided in which the housing comprises a one-piece block of metal bifurcated to form a pair of arms freely extending in opposition to each other and having at their free ends a circular loop bridging the arms and means for drawing the arms together to close the loop. The crank lever, to which the brake cable is attached is pivotally mounted between the arms and a lightweight tension spring provided to bias the crank lever into a rest position.

In the present invention, the arms extending in a cantilevered fashion spring at their base and, therefor can easily be drawn or pinched together evenly and uniformly. As a result, the loop clamp closes in a near perfect circle about the cylindrical handlebar, producing a sure clamping of the

assembly to the handle bar. Twisting or torquing forces created by the large tension of the brake cable are thus not transmitted to the clamp.

Preferably, the block from which the housing is formed is of a lightweight but strong and durable material such as aluminum. Thus, a strong housing can be machined, the strength of which can be enhanced by the means for drawing the arms together as well as mounting the crank lever. Set screws are preferably used to compress or draw together the arms. Still further, the use of such materials provides an extremely lightweight device, which may be further lightened by providing shaped relief areas in the body.

Full details of the present invention are set forth in the following description and illustrated in the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

Fig. **1** is a front perspective view of the hand brake of the present invention, illustrated as applied to the handlebar of a bicycle;

Fig. **2** is a rear perspective view of the hand brake shown in FIG. **1**; and

Fig. **3** is an exploded view of the hand brake of the present invention.

DESCRIPTION OF THE INVENTION

As seen in the figures, the hand brake lever assembly of the present invention comprises a body, generally depicted by the numeral **10**, having a base **20**; a bifurcated central portion **30** in which a pivotal lever **40** is mounted; and a clamp **50** bridging the bifurcations, by which the assembly is secured to the handlebar **12** of a bicycle.

Preferably, the body **10** fashioned from a single piece of rectilinearly shaped lightweight metal such as aluminum, titanium, or the like, which is also strong and highly non-malleable. The single piece of metal can be formed as a casting or can be worked as by cutting and shaping. The lever **40** is similarly preferably made from a single piece of metal such as that from which the body is formed.

As seen clearly from FIG. **3**, the base **20** is formed as a solid block extending forwardly of the central portion **30** and is provided with a through bore **22**, generally perpendicular to the lengthwise direction of the central portion **30**. The bore **22** is provided with a threaded surface **24** extending from the front face **26** of the body. The body **20** terminates in a wall **28** within the central portion **30**, this wall being smooth so as to provide an abutment surface for the lever **40**.

The central portion **30** comprises a pair of flat facing arms **32**, spaced from each other by a distance **34** and a length which will permit the lever **40** to move freely within it. The arms **32** have a thickness which prevents the arms **32** from bending and twisting under stress and torsion as applied during the braking operation but which, because of the cantilevered length of the arms, have a degree of springlike flexibility relative to the base **20**. At the upper end of each of the arms **32** but spaced from the beginning of the loops **50** are aligned holes **36**, through which passes a bolt **38** on which crank **40** is hung. The bolt **38** is secured by a suitable nut **39** and provided internally with suitable spacers to prevent the crank from engaging the inner surface of the arms.

The crank 40 comprises a bell-crank of the second order, having an elongated handle 42 and an angularly disposed beam 44. The beam 44 is provided with a hole 45 at its toe-or free end, which allows the crank to be freely placed over the bolt 38, thereby forming a fixed fulcrum for the crank. The outer corner 46, i.e. at the intersection of the handle 42 and beam 44, is formed with a shallow slot 47 parallel to the faces of the crank, which extends along the lower edge of the handle 42 to terminate in a circular opening 48 in one face of the handle. The opening 48 extends laterally through the handle and is so placed along the edge of the handle 42 that a radial opening 49 is developed.

Because the bell crank of the present invention is basically L-shaped, the line between the fulcrum and the corner 46 is generally perpendicular to the handle 42, and the shallow groove 47 lies at their intersection. As a result, the moment of turning of the cable about the elbow, defined by the groove 47, is optimum and the degree of force needed to pull the cable is minimum.

The attachment clamp 50 comprises a pair of loops 52 extending integrally from the arms 32 of the central portion, each arm 32 being thickened to form a pair of laterally spaced pedestals 54, which respectively form transversely aligned pairs across the space 34 between the arms 32, upon each of which a loop section 52 is formed. At the base or ends of each loop section 52, the respective pedestals are provided with threaded bores 56, into which threaded set screws 58 are placed, thereby enabling the loop sections 52 to be tightened on the handlebar 12, merely by drawing the pedestals closer to each other. Preferably, each pedestal is buttressed for strength and so that at least one buttress could act as a spacer or stop means, preventing the excessive tightening of the loop or the narrowing of the space 34 between the arms 32. The loops can thus be tightened into a near perfect circle.

Lastly, a tension spring 60, having loops 62 at each end is secured at one end over the forwardmost set screw 58 and at its other end to the forward edge of the beam 44 by a set screw 64 threaded in a hole 66.

The present device may be used with any conventional brake system using a cable—that is, with a brake system acting on the bicycle wheel hub or on the wheel rim. In placing the device into use, the brake cable 70 located in a sheath 72 (FIG. 3) is inserted through the conventional cable adjuster 74, which is screwed into the bore 22 in base 20. After pivoting the handle 42 to the position seen in FIG. 2, the free end of the cable 70 is wound about a threaded post 76, which is then seated into opening 48 and held in place by nut 78, which seats in the opposite face of the lever. The cable 70 enters the opening 48 through the edge opening 49 and seats itself within the shallow corner slot 47 of the crank beam 44.

The assembly may be fitted to the handlebar of the bicycle either before or after the cable 70 is attached to the crank 40. Preferably, the device is first placed loosely onto the handbar 12 by slipping the loops 52 over the end of the handlebar. The assembly is then located to the position desired by the biker and secured into place by tightening the set screws 58 below the loops 52. Although the buttresses generally provide fixed stops, care should be taken not to overtighten the screws 58, and they should be torqued evenly.

Lastly, the cable 70 is pulled taut so that the forward edge of the crank beam 44 abuts against the inner wall 28 of base 20. The brake end of the cable 70 (not shown) is then attached in normal fashion to the brake so that it remains taut while the brake is at rest or free wheeling position. Thereafter, pulling up on the handle 42 will activate the brake in a conventional manner. Release of the handle 42 will automatically result in the pivoting of the crank 40 into its rest position, being urged into that position by the tension spring 60 and the inherent tension of the cable 70. The spring 60, therefore, does not have to be of heavy weight or apply a great deal of tension. This contributes greatly to the lightness and ease of operation of the device.

It will also be seen that the brake lever assembly of the present invention is formed of only two major, integrally formed parts—namely, the body 10 and the crank lever 40. As each of these parts is preferably formed from a single block of metal, respectively, they are light and unencumbered by weldings, fastening members, connectors, and the like. Suitable machining results in a smooth, easily operable, streamlined, attractive design.

The use of aluminum such as 6061-T6 aluminum results in a brake handle assembly of approximately 50-55 grams per assembly, less than half the weight of conventional handle assemblies.

Various modifications and changes have been disclosed herein, and others will be apparent to those skilled in this art. Therefore, it is to be understood that the present disclosure is by way of illustrating and not limiting of the present invention.

What is claimed is:

1. A hand lever assembly for operating the cable of a bicycle brake system comprising a housing and a crank lever pivotally mounted on said housing and to which the cable of the bicycle brake is attached, said housing comprising an elongated monolithic block defining at its lower end an integral closed base and at its upper end an integral closed head, and having a slot formed in said block along its length between said base and said head end forming a pair of arms extending in spaced opposition to each other at a predetermined distance when said arms are drawn together and between which the crank lever is freely located, the head comprising a circular clamp having loop sections bridging and integral with both said arms, and forming with each arm a pair of laterally spaced, transversely aligned pedestals, each pedestal being buttressed with a transversely extending enlargement below the circular clamp, at least one of said buttressing enlargements extending transversely between the arms to thereby provide stop means for limiting the arms against being drawn in contact together, screw means extending through said respectively aligned pedestals for drawing said pedestals together to close said circular clamp when attaching the assembly to the bicycle and fulcrum means extending transversely between said arms below and spaced from said pedestals on which said crank lever is mounted, said crank lever being an L-shaped bell crank having a fulcrum hole at its free end and being fully pivoted on a bolt secured in said housing between the arms thereof.

2. The assembly according to claim 1, wherein the closed end of said housing is formed with a solid base from which said arms extend cantilevered and relative to which said arms are flexible.

3. The assembly according to claim 2, wherein said base is formed with a bore through which the cable passes, said bore extending perpendicular to the vertical plane.

4. The assembly according to claim 3, wherein said bore is threaded and adapted to receive a cable adjustment mechanism.

5. The assembly according to claim 1, wherein said housing is rectilinear and the bifurcation is made in the lengthwise direction, parallel to the corresponding faces to form planar opposing arms.

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